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Jet Flying-Boat Fighter Tests Resumed

THE major battles of the Korean war have emphasised the need for supporting ground and naval forces by fast, heavily armed fighters. One of the most important problems of modern warfare is how to bring air cover right over the scene of operations in the early stages of attack. Conventional types of fighter squadrons can be flown off advance airfields or aircraft carriers, but airfields have first to be captured and manned, and aircraft carriers brought close in to the battle area. The need for attacking aircraft which could operate from a stretch of sheltered water or from an inland bay has already been recognised in the war against Japan, and design work on a single-seater jet-propelled flying-boat fighter was begun at that time by Saunders-Roe. The Pacific war had come to an end some time before the SR/A1 had been completed, but work continued on three prototypes. Accidents to the first two models had brought trials to a standstill; however, tests with the third machine are now being carried out at Cowes, Isle of Wight.

The results of these new tests will be analysed and the rôle of the SR/A1 in combat more closely assessed. Design data have already been given to the U.S.A. under arrangements for the exchange of information.

What is Russia's Industrial Strength?

The memory of the highly creditable Soviet war production between 1941 and 1945, coupled with post-war economic growth and the possibilities of seizing the productive plants of Continental Western Europe, forbids complacency towards the potential military challenge of Russia's economy.

No one can estimate Russia's strength with any degree of accuracy, and the scanty information available is based on speculation, deduction, and the combination of little bits of news put together like a jig-saw puzzle. In view of this lack of reliable data, it was with great interest that we read of the impending publication of the book "Minerals—A Key to Soviet Power," extracts from which have recently been published in some American Journals. The author, D. B. Shimkin of the Russian Research Centre, Harward University, is to be congratulated on his unbiassed and matter-of-fact account based on years of study of the Soviet Union's resources.

The extracts before us deal with agricultural and mineral resources, the special problems of their distribution and transportation, and the economic developments which have taken place in the last decade, showing how the Soviets, through the utmost use of their capacity, have recovered from the severe losses of the War.

What have been the achievements of the fourth Five Year Plan to date? According to official Soviet statistics, industrial production, after a rise of 20 per cent in 1949 over the previous year, had exceeded the 1940 level by 41 per cent, a rate of progress in excess of the Plan.

Two questions arise from these claims: are they true? If so, how have the Soviets been able to achieve them? It is the author's opinion that the figures for increases in industrial production are inflated, and this belief is substantiated by the great disproportion between claimed advances in manufacture as opposed to consumption of raw materials. Nevertheless, it is clear that the country's industrial output in 1949 had actually exceeded the 1940 level by a substantial margin.

Despite the fact that world attention is focussed on Korea, it may not generally be known that Manchuria and North Korea would immensely strengthen Russia's position in Asia, since they form the only promising base for large-scale steel production east of the Ural Mountains. The differences between Russia's Eastern European and Far Eastern steel potentials are dealt with in great detail by the author, who supports his arguments with interesting maps and tables.

An Unusual Task for X-Rays

Industrial radiography has chiefly been used for the purpose of revealing blow-holes, porosity, inclusions, and other internal faults and discontinuities in materials. It has recently been shown, however, that X-rays can perform duties beyond the inspection of castings and welds. The Eastman Kodak Co. cites a case where two of a group of complicated hydraulic mechanisms failed to function properly. Stripping the whole mechanism would have been a costly procedure. It was suggested, therefore, that radiograms be taken of the faulty mechanism and that these should then be compared with a radiogram of a properly functioning unit. The comparison revealed that some springs were out of alignment and others were missing in the valve compartments. It is evident that this technique has great possibilities in diagnosing internal troubles in intricate mechanical installations, without the necessity of dismantling them.

Pulsating Jet Engines—A Survey of the Development of Ignition

By P. SCHMIDT. (From *Zeitschrift des VDI*, Vol. 92, No. 16, June 1, 1950, pp. 393-399, 14 illustrations.)

A CHARACTERISTIC feature of the pulsating jet engine is the automatic ignition which is propagated through the combustible mixture as a flame front of very high velocity. Work on the development of this type of ignition—about which little has been published—was started in 1931 in an attempt to solve the problem of pulsating high frequency combustion at constant volume in tubes open at one end. For this purpose, rates of flame propagation far in excess of those occurring in internal combustion engines were required.

Single ignition experiments were based on Wendlandt's finding that detonation of normally non-detonating gas mixtures could be induced by hard shock waves. A small volume of a volatile liquid fuel, such as ethyl ether, was added to the explosion chamber *a* (see Fig. 1), compressed air of 5 atmospheres was blown in, and the mixture was ignited by sparking plug *e*. The explosive blast fractured cellophane diaphragm *c*, initiating a shock wave in combustion tube *b* filled with the combustible mixture. Under suitable conditions, detonating combustion was obtained with a variety of fuels.

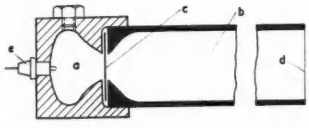


Fig. 1. **Equipment for shock wave experiments.**
(a) Explosion chamber ; (d) paper diaphragm ;
(b) combustion tube ; (e) sparking plug.
(c) cellophane diaphragm ;

The effect was similar when the initial shock—the bursting of the cellophane diaphragm—was caused by the introduction of a compressed non-burning gas mixture into chamber *a*. Although, on expansion, the temperature of the compressed gas tended to drop very low, ignition was obtained as long as the pressure had been above a critical value. Careful consideration led to the conclusion that detonation could have been caused only by the shock wave. The efficiency of energy transformation in these experiments with tubes open at one end was determined as 70 per cent.

In the next stage of development, single igniting shock waves were produced mechanically by a free piston arrangement. This was followed by the periodically operating device shown diagrammatically in Fig. 2. Ports *b* permit the combustible mixture to enter cylinder *a*. Free piston *e* moving to the right forces out the portion of the mixture not required for ignition through flap valves *d* into the combustion tube (not shown) to the right of the small opening *f*. On further movement of the piston to the right, the mixture trapped in the cylinder is compressed adiabatically to self-ignition temperature. The sudden rise in pressure forced a shock wave through opening *f*, causing the ignition in

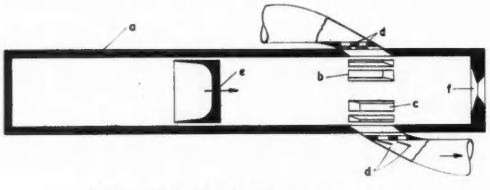


Fig. 2. **Pulsating ignition apparatus.**
(a) Cylinder ; (d) flap valves ;
(b) intake ports ; (e) free piston ;
(c) exhaust ports ; (f) opening.

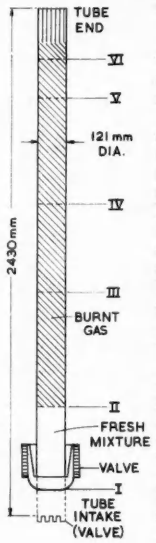


Fig. 3. **Cylindrical experimental tube with pressure pick-ups located at I to VI.**

the tube to the right of the opening, and also returned the piston to its initial position.

It was found in 1937 that in pulsating tube operation under suitable conditions, ignition was automatic, so that the apparatus for producing shock waves was no longer required. However, it was important for automatic pulsating ignition that the area of the combustion air intake and the sectional area of the exhaust end of the tube should be carefully related. With a tube shown diagrammatically in Fig. 3, pressure oscillograms were taken at several points. They are reproduced in Fig. 4. It can be seen that the explosive pressure wave (line *a*) is reflected back at the tube end as suction wave *b*. This depression is followed by a short, but very steep, pressure rise, which at point t_i hits the front of new combustible mixture, starting combustion immediately. A new small shock wave (*c*) is initiated.

The pulsating jet tubes used in these early experiments (described here for the first time) had only a

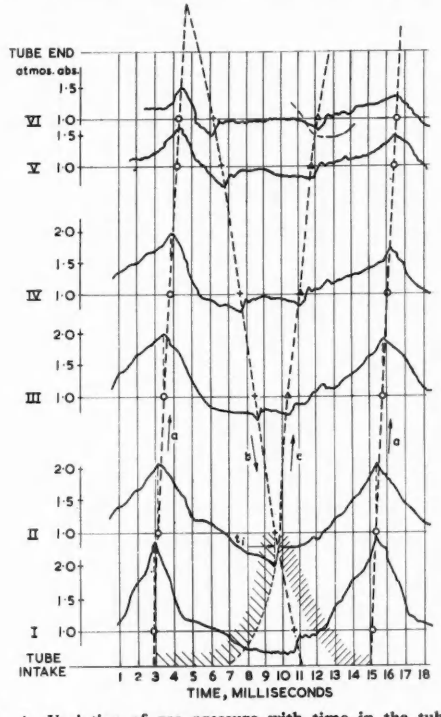


Fig. 4. **Variation of gas pressure with time in the tube shown in Fig. 3.**

moderate output, operating at maximum pressures between 2·5 and 3·5 atm. Later on, tubes of about 500 mm diameter operated continuously with maximum pressures of 5-6 atm. abs. The performance of these tubes was almost entirely independent of the fuel used. The tube diameter could be varied between 10 and 600 mm, and the shape of the tube, cylindrical or barrel type, straight or coiled, or the method of mixture formation, had no influence. Ignition was extremely reliable and was not stopped by considerable deformation of the tube or severe damage to valve or tube wall. Operation was not impaired by moderate static pressures at the exhaust end and can take place at considerable altitude.

U. S. A.

Rapid Tempering by Induction Heating

By J. F. LIBSCH and A. E. POWERS. (From *Metal Progress*, Vol. 58, No. 2, August 1950, pp. 176-180, 6 illustrations.)

Tempering of steels after quench-hardening is usually done slowly in a furnace, and the tempering temperature is maintained for some time, to improve penetration. This article shows that short-time tempering by induction heating is applied successfully to plain carbon and low-alloy steels.

INDUCTION heating is the quickest and most efficient way of generating heat in a metal piece. The required maximum temperature is reached at an almost constant heating rate, and not asymptotically as in conventional furnaces. Induction heating, therefore, offers considerable practical advantages for the tempering process if it can be shown that the tempering effect, the mechanical properties of the material, and the relief of hardening stresses are not adversely affected by the shortened cycle. The test specimens were heated with a Lepel high-frequency converter at about 300,000 cps. A chromel-alumel thermocouple (0·012 in. wires), cold-welded to the specimen and connected to a controller-recorder with electronic amplification, served to maintain the desired temperature at the specimen surface for any length of time, and to record the temperature on a pen-recorder with less than 10° F deviation even at the fastest heating rate. The following tests were conducted on steels tempered in various ways :—

(*a*) and (*b*) *Hardness and impact tests :* 0·450 in. diameter cantilever impact test specimens (ASTM type Y) with two standard notches at 1·125 in. distance were used, and the notches checked by optical comparator. Specimens of 3140 steel (0·4C ; 0·75 Mn ; 1·25 Ni ; 0·6 Cr average composition) were water-quenched after furnace-heating to 1550° F, and then cooled to—100° F in a dry-ice and alcohol mixture, so that any retained austenite would be transformed. Specimens of 1050 steel (0·5 C ; 0·75 Mn average composition) were brine-quenched after furnace-heating to 1575° F, and then cooled to—100° F. Both steels were finally tempered at intervals of 100° F, from 300 to 1200° F, the final temperature being maintained for one hour after furnace-heating, and for 5 or 60 seconds after induction-heating at a rate of 150° F per second. All the specimens were finally water-quenched after tempering to limit temper brittleness and provide precise holding times for the short tempering cycles. Rockwell hardness and cantilever impact tests were taken.

(*c*) *Tensile tests :* 0·505 in. diameter standard ASTM tensile test specimens of 3140 steel were heated to 1550° F, in a controlled-atmosphere furnace and quenched in water. Some specimens were then furnace-tempered for one hour at various temperatures, others induction-heated at a rate of 300° F per second and quenched immediately on, or 5 seconds after, reaching the required tempering temperature. Tensile stress-strain curves were taken on the tempered specimens, and the Brinell hardness ascertained at points well removed from the fracture.

(*d*) *Stress relief :* Three rings, 1·75 in. dia., $\frac{1}{8}$ in. thick, $\frac{1}{2}$ in. wide, made of 4130 steel (0·3 C ; 0·65 Mn ; 0·65 Cr ; 0·2 Mo average composition), were hardened by induction and slit radially. The gap width was measured by means of an optical comparator. The rings were then tempered at intervals of 100° F, from 200 to 1300° F, a heating rate of about 220° F per second being used. The change in gap width was measured for each tempering cycle. One specimen was induction tempered for 5 seconds, another for 60 seconds, at constant temperature, whilst the third test piece was tempered for 1 hour in a furnace. As a comparison, the Rockwell-C hardness was ascertained on similar specimens similarly treated.

(*e*) *Differential tempering :* Eight round cantilever impact test specimens, 1 in. diameter, 4·5 in. long, were machined from 4150 steel (0·5 C ; 0·75 Mn ; 0·95 Cr ; 0·2 Mo average composition) and provided with two 45 deg. notches machined to a depth of 0·075 in. with a bottom radius of 0·01 in. (same dimensions as for notches in 0·45 in. diameter specimens). The specimens were furnace-hardened by austenitizing at 1550° F, and quenching. Four specimens were then induction-heated at a rate of 80° F per second and water-quenched immediately on reaching the tempering temperatures. Four more specimens were furnace-heated and tempered for 1 hour at slightly lower temperatures to similar hardness figures. All the specimens were broken in impact tests, and the Vickers hardness was read at the centre and surface of each fracture (see Table I).

TABLE. I.—HARDNESS AND IMPACT ENERGY OF 1 IN. DIAMETER BARS OF 4150 STEEL, WATER-COOLED AFTER TEMPERING.

Specimen	Temper Treatment			Rockwell Hardness R_C		Impact Energy ft-lb. average
	Temp. °F	Time	Method	Surface average	Centre average	
1 and 2	1000	0 sec.	Induction	47·1	52·1	211
3 and 4	880	1 hr.	Furnace	47·8	47·8	214
5 and 6	1200	0 sec.	Induction	40·5	46·0	349
7 and 8	1040	1 hr.	Furnace	39·8	39·8	370

TEST RESULTS.

(*a*) *Effect of tempering time and temperature on hardness* (Fig. 1). In order to ensure uniform heating across the section, the Vickers hardness was read across the diameter of one specimen tempered 5 seconds at 1200° F. No variation in hardness was found. Even a 1-in. diameter specimen of 4150 steel, tempered 5 seconds at 1200° F after heating at 80° F per second, was uniformly hard throughout. Fig. 1 illustrates the similarity of results obtained by furnace and induction tempering if the reduced softening effect due to the short time of tempering is taken into account. Hollomon and Jaffe[1] have established the following time-temperature relationship for conventional methods of tempering :

$$R_C = H_C - 0{\cdot}00254\,T\,(13{\cdot}0 + \log t),$$

where R_C is the actual Rockwell C hardness, H_C an extrapolated hypothetical hardness figure, T the temperature, t the time. A comparison of values calculated from this formula with the experimental results of Fig. 1

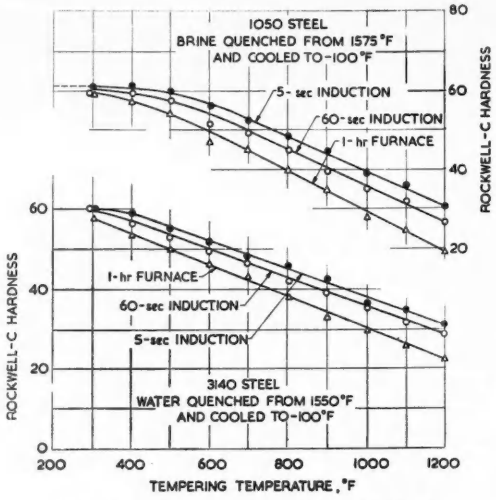


Fig. 1. **Tempering curves for 1050 and 3140 Steels.**

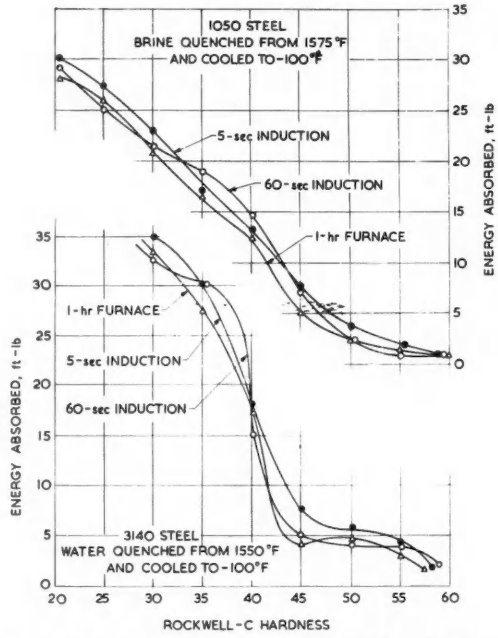


Fig. 2. **Impact energy versus hardness for 1050 and 3140 Steels, tempered to similar hardness by various tempering cycles.**

shows that this formula can be extended to hardness figures obtained by induction-tempering. Important deviations only occur at tempering temperatures above 1000° F.

(*b*) *Impact properties* (Fig. 2). The impact tests at room temperature show that, within experimental errors, the impact values are a function of the hardness, regardless of the duration of tempering. As temper brittleness is caused by precipitation of a phase[2] with maximum reaction at 950° F, it may be thought possible that short-time tempering avoids temper brittleness. However, the curves of Fig. 2 lead to no definite conclusion. Since impact tests at room temperature do not provide any satisfactory criterion for temper brittleness, further tests will be needed to study this possible effect.

(*c*) *Other mechanical properties* (Fig. 3). Since the hardness of induction-tempered specimens complies with Hollomon and Jaffe's formula, it may be expected that the conventional relation between Brinell hardness and the other mechanical properties is also valid, provided the steel is quenched to a fully martensitic

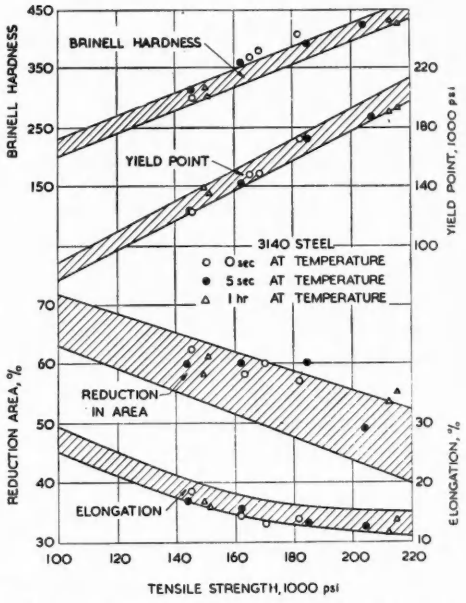


Fig. 3. **Correlation of mechanical properties of tempered 3140 Steel with most probable values found by W. G. Patton.**

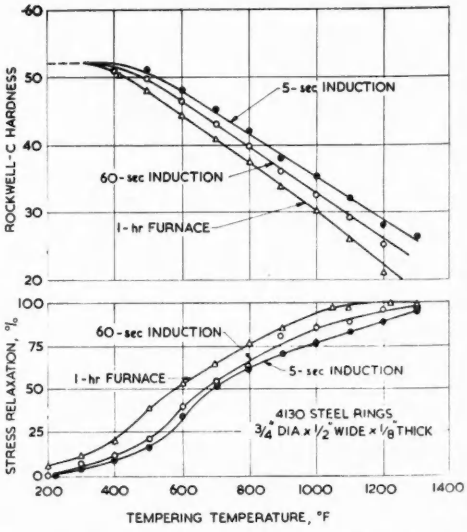


Fig. 4. **Stress relaxation and hardness for 4130 Steel rings hardened by induction and then slit.**

structure and tempered to below 200,000 psi tensile strength. Fig. 3 shows indeed that results lie within the ranges of the " most probable values " for tempered martensite, as established by Patton[3].

(*d*) *Relief of residual stresses* (Fig. 4). The change in gap width of the slit rings, before and after a tempering cycle, may be taken as an indication of the relief of residual stresses due to hardening. A tempering treatment of 2 hours at 1200° F resulted in complete stress relief, i.e., no further change of gap width could be detected during continued tempering. The effects of shorter tempering treatments are plotted as percentages of this complete relief. It is seen (Fig. 4) that tempering for 1 hour at 1100° F already affords complete relief. Induction-tempering for 5 or 60 seconds does not produce 100 per cent stress relaxation at any temperature below the transformation range, although values of 96 and 99 per cent are reached at 1300° F. If stress relaxation is plotted against Rockwell-C hardness from Fig. 4, the three curves coincide within experimental errors, so that even for induction-tempered specimens, the hardness may be used as a practical measure of the stress relief effected.

(*e*) *Size effect* (Table I). Since the induced current has a maximum intensity at the surface of the specimen and is zero at the centre, a certain hardness gradient can be expected after tempering. However, the heat flow inside the specimen is so rapid that the Rockwell-C hardness of a 1 in.-diameter specimen heated at 80° F per second and quenched immediately on reaching tempering temperature is only 5 points higher at the centre than at the surface. If the bar is held 5 seconds at the tempering temperature, no difference at all is noticed. It may be concluded that, in the first case, a maximum temperature difference of 150° F existed between surface and centre. In spite of this gradient, the impact strength is equal for induction-tempered and furnace-tempered specimens of the same surface hardness (Table I). More rapid heating rates and larger cross-sections would be required for a marked surface-tempering effect.

Conclusions.

Induction heating for short-time tempering appears practical provided the temperature is increased to compensate for the short time. The hardness obtained by a given tempering cycle can generally be predicted from Hollomon and Jaffe's formula. The mechanical properties (impact and tensile tests) as well as the relief of residual hardening stresses can generally be ascertained from the tempered hardness irrespective of the tempering cycle. The hardness figure can, therefore, be used to control the induction-tempering operation. Unless very rapid heating rates and large cross-sections are used with a quench immediately on reaching the tempering temperature, the hardness gradient across the section of a tempered specimen has no importance and the impact strength is determined by the surface hardness only. A surface temper with a hard core appears feasible under suitable conditions.

REFERENCES.

(1) Hollomon and Jaffe, Time-temperature relations in tempering steel, *Trans. A.I.M.E.*, Vol. 162, 1945, p. 223.

(2) Greaves and Jones, Temper brittleness of Ni-Cr-Steels, *Journal Iron & Steel Inst.*, Vol. 102, 1920, p. 171.

(3) W. G. Patton, *Metal Progress*, June 1944.

SWEDEN

Gas-Sampling Apparatus

By T. Wilner. (From *Teknisk Tidskrift*, Vol. 80, No. 23, June 10, 1950, pp. 553-554, 3 illustrations.)

The apparatus described in this article was originally developed by the Swedish Defence Research Station for the detection of very small quantities of war gases in the air. It is also particularly suitable, however, for rapid determination of the concentration in air of halogen substitutes such as carbon tetrachloride, trichlorethylene, chlorobenzene and cooling agents of the freon type, etc. The apparatus, shown in Fig. 1, is a light and simple design. An electrically driven miniature fan forces the air to be tested through a small catalyser tube in which the substance being investigated is decomposed by contact with an electrically heated platinum spiral. The acid products of decomposition impart a strong red colouring to a specially impregnated reagent paper. The small-size apparatus uses very little electrical energy, with an operating voltage of 5 volts and a current of 0·3 ampere. The reagent paper is used dry.

Maximum sensitivity is obtained by keeping the paper in a container with a drying agent (blue gel), and removing it only immediately before the test. The colouring effect is reversible, that is, the red colour disappears fairly rapidly after the test is completed. The paper shows no reaction to substances of low acid content, the result being that a number of gases which may have dangerous effects are not detected. Substances containing chlorine can, however, be detected in concentrations as low as 1 milligramme per cubic metre of air. A reaction is already noticeable if quantities as small as 10^{-7} to 10^{-8} gram pass through the catalyser tube. For halogen-free sulphurous substances the sensitivity is lower, partly because the products of decomposition are not effectively absorbed by the reagent paper.

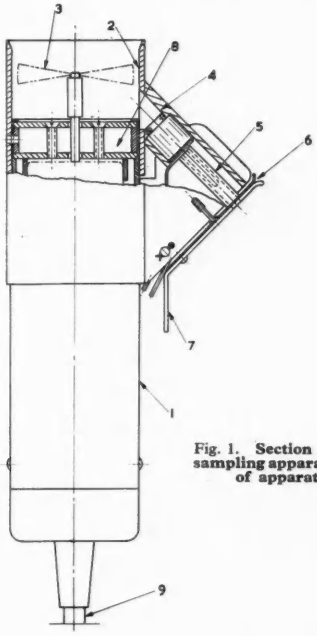


Fig. 1. **Section through the gas sampling apparatus. Total length of apparatus 4¼ in.**

The construction is shown in detail in Fig. 1. The cylindrical case (1) contains a small electric motor, which drives the fan (3) enclosed in the removable fan housing (2). This housing has an obliquely arranged cylinder (4) containing the easily replaceable catalyser tube (5). The paper is placed in the flexible holding member (6) at right angles and directly against, the tube orifice, which is ground flat and provided with two cross-channels, through which the combustion products flow out and come into direct contact with the paper. The trace of the reaction on the paper thus has the shape of a cross and can be easily observed during the test by pressing lightly on flap (7) and lifting the paper off the tube orifice for brief inspection. For a prolonged test, in order to obtain, for instance, a reaction of greater intensity, care must be taken to avoid shifting the paper in its holder. The paper can be removed by pressing at the bottom end of flap (7) which opens the clamps holding the paper. A thin oil layer formed around the motor shaft can absorb halogenous gases. To avoid this difficulty, an air chamber (8) is arranged between the fan housing and the motor. A slight air current is therefore made to pass around the shaft and flows out through a lateral hole. This arrangement eliminates disturbing effects which might otherwise occur during prolonged exposures for intensive reactions. The battery leads (9) are provided with a plug-in contact for connection to a light accumulator in a portable case.

For concentration determination, a measurement is taken of the time which has elapsed until a colouring of a given intensity is obtained on the reagent paper, using a standard colour table with different colour intensities. The following table shows examples of short-time tests with trichlorethylene and paradichlorobenzene, for which the colour intensities obtained were compared with those normally obtained for prolonged exposures. The time values have been rounded off to the nearest second.

	Concentration (mg/m³)				
	50	100	200	250	500
Trichloroethylene	–	2	1	–	0
Paradichlorobenzene	8	5	–	2	1

The apparatus can also be used to check the airtightness of refrigerating equipment using chlorine-type cooling agents, such as freon or methyl chloride. The high sensitivity of the apparatus makes it possible to obtain a general check of an entire installation by means of a single test. For this purpose, the apparatus is taken into the machinery room or, in some cases, placed in the ventilation ducts communicating with the machines. If no reaction is observed, it may be concluded that

(*Concluded on page* 408)

ITALY

Problems of Life, Efficiency, and Colour in the use of Fluorescent Lamps

(From *Quaderni di Studi e Notizie*, Vol. 6, No. 76, June 1, 1950, pp. 395-400, 6 illustrations.)

The rapid progress made in recent years in electric lamp manufacturing technique has resulted in a number of intensive investigations being simultaneously carried out to determine which of the many sources available would be the most favourable for any individual lighting problem in industry, commerce and the home.

INSUFFICIENT knowledge on the part of the contractor or user of fluorescent lamps is most likely to lead to an incorrect choice and, consequently, inferior results. The fact that the number of users of fluorescent lighting is steadily growing only adds to the importance of this problem. After having found wide and rapid acceptance for industrial and commercial lighting, fluorescent lamps are being used more and more in other fields, such as in street lighting and in the home. In the United States, for instance, the number of homes provided with flourescent lamps has doubled in the past four years, with the result that in 1949 such lamps could be found in 19 per cent of all homes.

TABLE I.—SPREAD OF FLUORESCENT LIGHTING IN HOMES IN THE U.S.A.

Year	Homes equipped with fluorescent lighting	Rooms of homes equipped with fluorescent lighting					
		Kitchens	Sitting rooms	Dining rooms	Bathrooms	Bedrooms	Other rooms
	in per cent of total						
1945	10·2	7·6	2·5	1·1	0·7	—	1·5
1947	15·7	13·0	3·0	1·1	—	—	2·1
1949	18·9	13·5	3·1	1·1	2·3	1·1	2·3

Considering this trend of development, it may be useful to review a number of essential questions concerning this type of lighting and, in particular, to deal with the differences in the various types of fluorescent lamps at present on the market from the viewpoint of their useful life and luminous efficiency, and also to review the general problem of using the various possible shades of colour obtainable with fluorescent lighting.

Three types of fluorescent lamps

Fluorescent lamps being produced at present on an industrial scale can be roughly divided into three groups. In the chronological order of their appearance on the market, these are " cold-cathode lamps," " hot-cathode lamps " with preheat starting, and, finally, hot-cathode lamps with instantaneous starting. The latter, which first appeared on the American market only four years ago, are sometimes known as " slimline " lamps. In addition, there exists, of course, a large number of other types, some of them designed for special purposes. Thus, there are lamps with instantaneous starting which, as regards dimensions, current and voltage, are similar to hot-cathode lamps with pre-heat starting ; these are generally made specifically for a capacity of 40 watts and require a special circuit arrangement. Then there are special lamps of small capacity (of the order of 1W) built in the shape of a bulb, which can be connected directly to the mains and produce immediate ignition ; they are used for signalling and similar purposes and have a useful life of about 10,000 hours. There are lamps for operation on direct current, lamps for low temperatures, and many others.

The three main classes of fluorescent lamps differ fundamentally as regards their *electrical* behaviour. Thus, to be able to assess their respective operational qualities, it will be useful first to describe briefly their behaviour during the starting-up period as well as during subsequent operation.

In the case of the cold-cathode lamps, the electrical discharge sets in immediately on application of the starting voltage, which varies, according to the operating conditions, between 2·5 and 4 times the normal operating voltage. During regular operation the discharge has the properties of a " cold discharge," characterised by a low current intensity, generally less than one tenth of an ampere, and by a high cathode voltage drop around 100 V); under normal conditions, the temperature of the cathode remains of the order of 150° C.

On the other hand, in the case of hot-cathode lamps, with pre-heat starting, the discharge does not take place until the cathodes have been heated to about 900° C, a temperature which is maintained throughout the operation of the lamp. This temperature is obtained in the first place by means of an auxiliary circuit during the " heating-up period," which may range from a fraction of a second to about 2 seconds. The discharge obtained in this case has the characteristics of an electric arc: a current intensity of several tenths of an ampere, and a cathode drop of less than 20 volts.

Finally, the electrical characteristics of " slimline " lamps are a combination of the characteristics of the two first-mentioned types. The starting-up is similar to that of the cold-cathode type, while, during operation, the temperature of the cathode is raised and the discharge assumes the characteristics of the " hot-cathode " lamp.

It is only natural that the different principles of operation should lead to different structural designs. The cold-cathode tubes are normally of small diameter and greater length, which, in the case of lamps of over 60 W capacity, may even exceed 3 metres; there is only one terminal at each end, and the cathodes appear robust and compact.

In the case of the hot-cathode lamps with pre-heat starting, on the other hand, the ratio of diameter to length is greater; there are two terminals at each end for connecting the heater circuit, and the cathodes consist of fine heater spirals. Finally, the " slimline " lamps have dimensions intermediate between the hot-cathode and the cold-cathode lamp; in common with the cold-cathode lamps, they have only one terminal at each end, while their cathodes consist of relatively large spirals. The operating voltage is also intermediate between those of hot-cathode and cold-cathode lamps, and normally lies between 130 and 340 volts.

Equally considerable differences are found in the operating characteristics of the three types, for example, as regards brightness, life and luminous efficiency.

LIFE

It is known to be one of the advantages of fluorescent lamps, as compared with incandescent lamps, that they have a much longer life. Without wishing to analyse the different meanings which the term " life " can assume in fluorescent lamps, it may be sufficient to say that it is a function of the inter-action of a large number of complex interdependent factors, all of which are more or less directly influenced by the conservation of the electrodes. These factors depend, on the one hand, on the design characteristics of the lamps (including, for instance, the active materials with which the electrodes are coated and the nature and pressure of the inert gas with which the lamp is filled), and, on the other, on purely external conditions such as the wave form and the level of the supply voltage and, in particular in the case of the usual hot-cathode lamps, the ambient temperature and the frequency of switching. As will be seen from Fig. 1, the life (expressed as the average number of hours of operation) increases noticeably, in the case of hot-cathode lamps, with the increase of the average duration of each period of operation. Quantitatively, the life of lamps built about two years ago varies, for instance, from 2,500 hours for individual periods of operation of three hours each, to 4,000 hours for periods of six hours and 6,000 hours for periods of twelve hours.

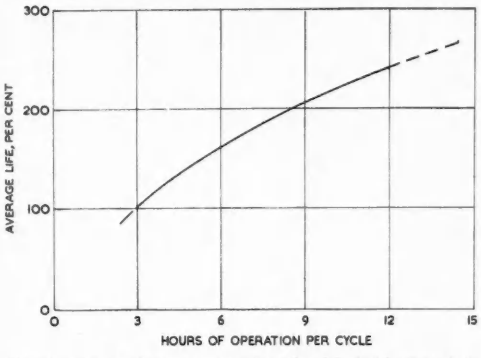


Fig. 1. **Effect of the average length of individual periods of operation on the total life of hot-cathode fluorescent lamps.**

These values for the life of hot-cathode lamps have been largely superseded today (see Table II). According to announcements made by the leading manufacturers it appears, in fact, that the most recent lamps will have a life of practically three times that mentioned above, and a value of 7,500 hours can be reached if the lamp is operated in individual periods of three hours.

TABLE II.—AVERAGE LIFE OF HOT-CATHODE FLUORESCENT LAMPS FROM 1938-1949. (Hours of actual operation).

Year	6 W	8 W	14 W	15-30 W	40 W	100 W
1938	—	—	—	1250	—	—
1939	—	—	—	2000	2000	—
1940	750	—	1500	2500	2500	2000
1941	750	750	1500	2500	2500	2500
1942	1000	750	1500	2500	2500	3000
1945	1500	1500	2000	2500	2500	3000
1947	2500	2500	2500	2500	2500	3000
1949	7000	7000	7000	7000	7000	7500

From the point of view of life, slim-line lamps behave similarly to lamps with preheat starting. They, too, are influenced *inter alia* by the number of operations and, in this respect, have developed at the same rate as the normal hot-cathode lamps with preheat starting.

On the other hand, the behaviour of the cold-cathode lamps as regards their life is entirely different from that of the lamps previously mentioned. Their life is practically unaffected by either the number of operations or (within wide limits) by reduced operating voltage. In actual operation, they have a life of more than 15,000 hours.

LUMINOUS EFFICIENCY

Directly connected with the characteristics which determine the life of fluorescent lamps is a large number of interesting photometric characteristics ranging from the luminous efficiency to the changes which this undergoes during the course of the life of the lamp.

As mentioned before, the merit of the whole range of fluorescent lamps is essentially based on their higher initial efficiency (see Table III), which in recent years has shown a rapid increase. However, the secondary physical and chemical phenomena which take place

TABLE III.—LUMINOUS EFFICIENCY OF HOT-CATHODE FLUORESCENT LAMPS FROM 1938 TO 1949 (Lumen/Watt).

Year	6 W	8 W	15 W	20 W	30 W	40 W	100 W
1938	—	—	32	32	35	—	—
1939	—	—	36	39·5	45	48·5	—
1940	30	—	39	45	48	53	—
1945	32	40	39	46	48	58	42
1946	35	41	42	48	50	58	44
1949	35	41	42	48	50	58	—

inside the fluorescent tube when in use, and in which the cathodes, the gas and the fluorescent powders are involved in various degrees, always produce a certain amount of deterioration of the luminous efficiency; also, the results obtained in extending the life of the lamps would today have no practical value had they not been accompanied by corresponding improvements in the extension of the period of high luminous efficiency. This made it possible, at least for hot-cathode lamps, to avoid excessive reduction of the efficiency making the *economic* life of the lamp shorter than its *effective* life.

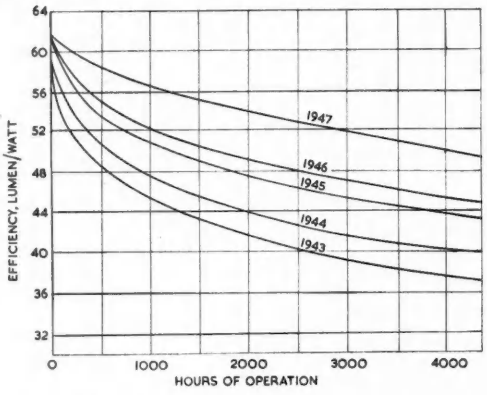


Fig. 2. **Effect of total number of hours of operation on the luminous efficiency of fluorescent lamps. Comparison of average values for lamps manufactured between 1943 and 1947.**

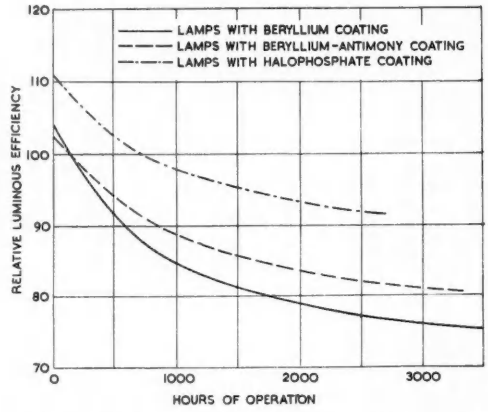


Fig. 3. **Effect of total number of hours of operation on the luminous efficiency of fluorescent lamps. Comparison of lamps with various phosphors based on an efficiency of 100 per cent for beryllium coated lamps after 100 hours of operation.**

The normal photometric behaviour of a fluorescent lamp during the course of its life can briefly be subdivided into three periods :—

(a) Initial period of about 100 hours, during which a relatively rapid decrease of the luminous efficiency can be observed, which is probably due to a chemical reaction between the fluorescent powder and the impurities of the gas contained in the tube.

(b) A second period of slow deterioration and longer duration (from 1,000 to 2,000 hours), during which chemical reactions take place between the fluorescent powder and the glass.

(c) A third and last period of very slow deterioration of the phosphors.

The progress achieved in the field of luminous efficiency which is due *inter alia* to the development of new types of fluorescent powders, is illustrated by the diagrams shown in Figs. 2 and 3, which show, year by year, a comparison of the mean values of the preservation of the luminous efficiency, known as "lumen maintenance curves" of lamps built during and after 1943. It can be said that the initial decrease of efficiency (during the first 100 hours of operation) is less than 10 per cent, and that, on average, it does not decrease during the 6,000 hours of operation to below 25 to 30 per cent.

THE SMASHING POINT

The behaviour of the lumen maintenance, which is essentially determined by the photo-emissive properties rather than by the electrical properties of the lamp, is more or less equal both for hot-cathode and cold-cathode lamps, provided the materials used are the same and the current densities of the discharge are equal.

However, this property has gained particular importance as regards cold-cathode lamps, as, due to their relatively long life, their luminous efficiency eventually decreases to a point where the replacement of the lamp becomes economically advantageous, although the lamp itself may still function. This limit of economic life, known in America as the "smashing point," obviously marks the number of hours of operation for which the minimum cost per lumen-hour produced has been reached. If the number of hours of operation is increased beyond this point, the cost of energy per lumen-hour increases due to the deterioration of the efficiency (Fig. 4, curve (A)), while, on the other hand, the charges of the cost of the lamp decrease, curve (B). The total cost of each lumen-hour produced, which is the sum of these two items, is shown in curve (C) as a function of the hours of operation. The number of hours of operation at which the minimum occurs is determined by the relative importance of the cost of electricity and the cost of the lamp.

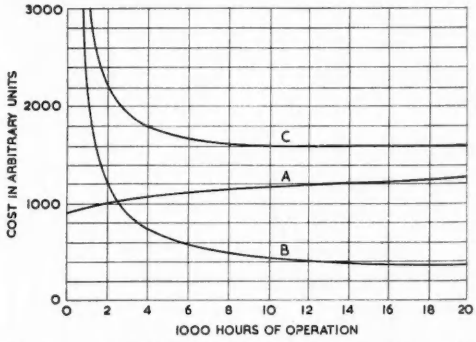


Fig. 4. **Cost per lumen-hour as a function of the total number of hours of operation of a cold-cathode lamp.**
Curve A: cost of electricity
Curve B: cost of replacement of lamp
Curve C: total cost per lumen-hour produced.

The particular curves shown in Fig. 4 were derived for a 75 watt cold-cathode lamp of a type, the luminous flux of which, initially 2,400 lumen, decreases by 20 per cent after 5,000 hours of operation and by 30 per cent after 15,000 hours. The cost of changing a lamp includes the cost of the lamp itself as well as the cost of labour. The minimum cost per lumen-hour is reached after an operating period of 14,000 hours. This minimum point, however, varies according to the cost of electricty.

COLOURS

The problem of general importance relating to all types of fluorescent lamps is to determine the most favourable colour of light in respect to the various possible applications. This involves numerous problems, both of a technical and physiological character; it may, however, be sufficient here to survey only the fundamental requirements which, based on recent experimental and statistical studies, should ensure a rational use of the new sources of light. A few additional notes will describe the practical solutions which the fluorescent lamp manufacturing industry is today able to offer.

Leaving aside colour illumination of an essentially decorative character as well as the lighting of open spaces, the requirements of tone in white interior lighting are determined mainly by the following factors:

(a) The purpose for which the illuminated space is required.
(b) The fundamental colour of the objects to be illuminated.
(c) Subjective factors connected with climate, habits, etc.

As regards the first item, a distinction can be drawn between two classes of interiors:—

On the one hand, interiors in which activities of a social nature take place, such as theatres, restaurants, living rooms and dining rooms; on the other hand, interiors designed for "occupational" and other work, such as offices, schools, factories, all kinds of shops, kitchens, etc. For the first group, tones of warm white colour are preferred, while colder tones are preferred in general for the second group.

As regards the second item, systematic researches have shown the existence of a correlation between the tone of the artificial light which falls on one object, and the natural colour of the latter, insofar as the sensation of the colour is more pleasing, the more the spectral curve of the artificial light approaches the curve of the light reflected by the illuminated object.

It must be noted, however, that the subjective preferences of tone of artificial light are also influenced by other factors, as, for instance, the climate, so that a cold tone seems to be more suitable for hot climates, and *vice versa*.

The fluorescent lamp industry can now meet these various requirements by providing an extensive range of tones of white light, which can be divided into five now standardised types:—

(1) White of about 6,500° K (daylight)
(2) White of about 4,500° K (brilliant white)
(3) White of about 3,500° K (sunlight)
(4) Soft white
(5) Warm white of about 2,800 to 2,900° K.

The first three tones, from the coldest daylight to sunlight, created with the intention of obtaining artificial light which would reproduce as best as possible the spectral composition of natural light, are used mainly for the lighting of work rooms and for general lighting. Thus, for instance, the white light of 4,500° K is particularly suitable for school room and shop lighting.

On the other hand, the last two tones of warm white and, in particular, the warm white (5) tend to approximate to the tones of incandescent lamps. These are primarily suitable for clubs and homes and offer a more satisfactory solution to the difficult problem of artificial lighting for food and for the human complexion. In conformity with the requirements described above and illustrated by an example in Fig. 5, the spectral distribution of this warm light is, in fact, similar to the mean curve of reflection of certain foods and of the human complexion.

This important aspect of fluorescent lamp technique is today undergoing full evolution. It may be of interest to mention as an example in this connection

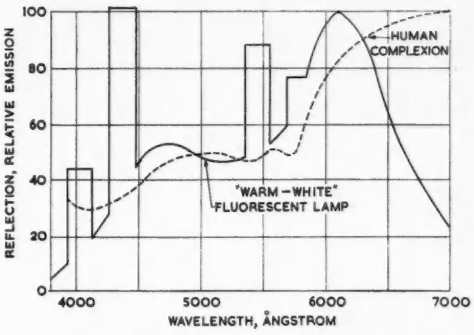


Fig. 5. Average spectral curve of reflection of the human complexion and the emission characteristic of a "warm white" fluorescent lamp.

The discontinuities in the emission spectrum of the lamp are due to the spectral lines of the mercury vapour contained in the tube which are superimposed on the continuous spectrum of the phosphors.

that a large manufacturing firm has only recently announced its intention of putting on the market this year two new series of lamps, which, due to the introduction of improved fluorescent powders (the so-called Deep Red, or "D R" phosphors) will be better equipped to produce any colour. According to this announcement, all problems of colour could be resolved by the combination of only four series of lamps, namely, the two already existing types of brilliant white of 4,500° K, and "warm white," and of two new types which, although of lower luminous efficiency, will make it possible to obtain a higher chromatic efficiency.

The preceding notes on the characteristics of operation of some fluorescent lamps have shown that the variety of types now in existence can enable the user to solve the most varied problems which present themselves in practice. On the one hand, a range of light tones is available which is adequate for any type of space and, on the other, there is in existence a series of types of lamps, ranging from cold cathode lamps to the "slimline" type, each of which has particular characteristics suiting it for a particular field of application.

The cold-cathode lamp, for instance, by its mode of starting, which within certain limits is not affected by low temperatures, is adapted for installation in the open. It can also be used in cases where frequent starting is called for (as this practically does not affect its life), or where gradual dimming may be required. Due to its long life, it is specially suitable where frequent replacement would be particularly inconvenient and expensive. Finally, due to its lesser brightness, which makes shading unnecessary, and because of its slim appearance, which makes it useful for architectural applications, it is particularly valuable for decorative lighting.

The normal hot-cathode lamp with preheat starting, on the other hand, which, due to its low operating voltage does not require special precautions for installation and operation, possesses a higher luminous efficiency, and the first cost of which is relatively small, finds its most suitable use in normal interior lighting. As regards the lighting of homes, in particular, its increasing use is more justified in the kitchen and in drawing rooms, where usually longer cycles of operation are needed.

The "slimline" lamp, finally, due to its higher efficiency, simplicity, and safety of installation and maintenance, equal to those of the normal hot-cathode lamp, also possesses the advantages of instantaneous starting and the elegance and decorative appearance of the cold-cathode lamp. This last type, therefore, is today tending to capture many fields of application, particularly in America.

GERMANY

A Platinum Resistance Thermometer for temperatures up to 750° C

By A. BARTHEL. (From *VDI Zeitschrift*, Vol. 92, No. 25, September 1, 1950, pp. 726-728, 7 illustrations.)

Platinum resistance thermometers are widely used in industry. Their effective range was hitherto restricted to an upper limit of 550° C, owing to the softening of the hard-glass resistance core at this temperature. By the use of special materials—sinter alumina and enamels—the range has been extended to 750° C.

ELECTRICAL resistance thermometers are employed in the range —200 to +550° C and have the advantage of high accuracy and easy readability at a distance. They consist principally of a temperature-sensitive resistance coil, an indicating instrument, a d.c. source and connections (Fig. 1). The indicator is usually a crossed-coil instrument directly calibrated in centigrade units. The temperature range of the thermometer depends on the materials used in the construction of the resistance coil, core and protective sheathing. In present-day practice, only pure platinum or nickel is used as resistance wire, because of their reproducible temperature characteristic, the average value of their specific resistance, and their freedom from corrosion. Nickel is used in the range —200 to +150° C, while platinum is used up to 550° C.

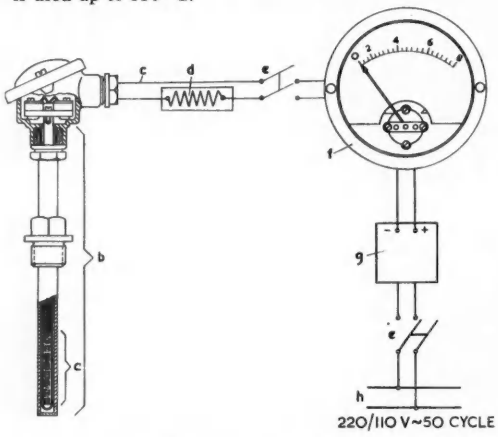


Fig. 1. **Thermometer set-up.**
(a) Resistance element; (b) complete resistance thermometer; (c) connecting leads; (d) compensating resistance; (e) switch; (f) crossed-coil instrument; (g) rectifier; (h) a.c. mains.

The measuring head hitherto chiefly employed for industrial purposes consists of a platinum winding on a hard-glass core, the whole sealed into glass. This arrangement has the advantages of sturdiness, freedom from the effects of corrosion, and good thermal conductivity, but owing to the softening of the glass at 580° C, it could not operate above 550° C. This was a considerable limitation on its usefulness, e.g., in the chemical industry. Recent work by the German firm of Degussa resulted in a resistance thermometer with an upper limit of 750° C.

Construction. The platinum resistance wire is doubly wound over a sinter alumina core (Fig. 2). The external connections are led through two channels and their contact with the resistance coil is secured by a sinter alumina bead. The protecting enamel layer extends over the bead, thus firmly holding the assembly together. The enamel layer has the same expansion coefficient as the sinter alumina core, and is an excellent insulator.

Dimensions. The thermometer is at present manufactured in the size indicated in Fig. 2, which is also the normal size of the hard-glass thermometers.

Electrical data. The temperature dependence of the pure platinum resistance R is given by $R = R_0 (1 + at + bt^2)$, where t = temperature in °C, R_0 = resistance at 0° C. The factors a and b are determined by resistance measurements at 0, 100 and 444·4° C (boiling point of sulphur). The temperature characteristic is altered by the presence of the covering layer, and obeys an equation of higher degree. In practice, the instrument is calibrated by means of a large number of fixed points on the temperature scale (usually melting points of pure metals). The temperature coefficient of resistance per ° C between 0 and 100° C is $3{\cdot}85 \times 10^{-3}$, the same as for the hard-glass thermometer.

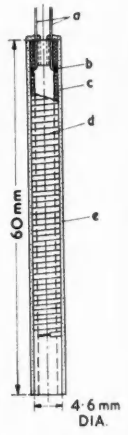


Fig. 2. **Resistance element of the 750° C thermometer.**
(a) Leads; (b) soldered joint in recess; (c) sinter alumina resistance core; (d) platinum resistance wire (0·05 mm dia.); (e) enamel layer.

Duration tests at the limiting temperature. Duration tests extending over many weeks were carried out at 600, 700 and 750° C. The thermometers were placed in a block of chrome-nickel and heated in an electric furnace. The resistances were measured at intervals, and were tested from time to time over the whole measuring range of —200 to +750° C. The results of these tests are shown in Fig. 3. The experimental readings had a scatter of ± 0·15 per cent, well within the required limit of accuracy. Duration tests at 800° C, however, already showed scatter outside this limit; hence, 750° C was fixed as the maximum operating temperature.

Hysteresis. Differences of the order of 0·10 ohm are observed in resistance measurements at increasing and decreasing temperatures. This effect is termed "hysteresis," and is of the same order with hard-glass thermometers.

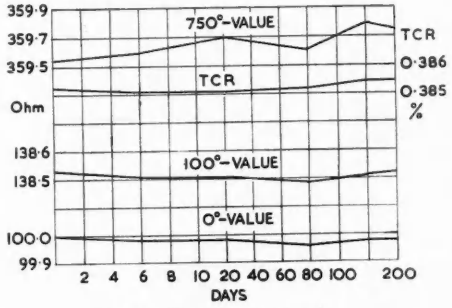


Fig. 3. **Duration test graphs.**
Typical result of a number of tests, showing the temperature coefficient of resistance (TCR), and the values of resistance at 0, 100 and 750° C.

Polarisation. On switching the energising current on or off, a current-pulse is observed. This so-called "polarisation effect" depends on the conductivity of the enamel and the resistance core. With hard-glass thermometers, this effect becomes troublesome at about 300° C, while it only appears at about 700° C and has no disturbing effects below 750° C with the new type of thermometer.

Speed of response. Fig. 4 shows that the response curve of the new thermometer is somewhat more favourable than that of the hard-glass thermometer.

Internal heating. The energising current heats the resistance coil to a greater or lesser extent, depending on the thermal conductivities of the materials employed, the sheathing and the surrounding media. The internal heating is less if the conductivity of the medium is high. The effect, which has to be allowed for, is about the same for the two types of thermometer.

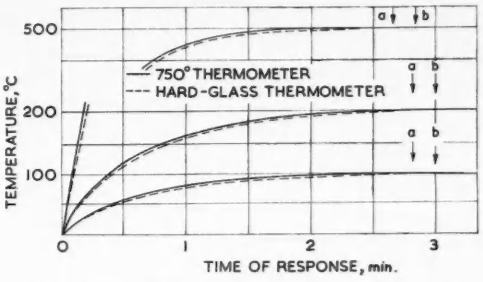


Fig. 4. **Response time curves of resistance thermometers.** The arrows indicate when the final temperature is reached. *a* = 750° thermometer; *b* = hard-glass thermometer.

AUSTRIA

Calculation of Stresses in Connecting Rods

By H. PARKUS. (From *Maschinenbau und Wärmewirtschaft*, Vol. 5, No. 9, September 1950, pp. 152-157, 5 illustrations.)

Connecting rod stress-calculations hitherto involved rather rough approximations. A more exact treatment requires a sufficiently accurate knowledge of the forces acting on the rod, from which the stresses should be calculable. In the present state of the theory of elasticity, it is still necessary to calculate nominal stresses at the weakest cross-sections, and to multiply these by theoretically or experimentally established stress-concentration factors.

FORCES ACTING ON THE ROD

THE force acting on an element of a moving body can be represented as the sum of the external forces and the d'Alembert inertia forces. The rod is guided in its path by the crankshaft and the piston, and thus, in addition to its own inertia forces, a load is exerted by these adjoining elements. The angular volocity ω of the crankshaft may be assumed to be constant.

The inertia forces exerted by the piston can be written down immediately, using the known expression for the acceleration of the piston, and the same applies to the forces due to the acceleration of the rod. It remains to calculate the inertia and Coriolis forces due to the relative motion of the two. However, simple consideration shows that the latter can be neglected. For, if ν be the frequency of oscillation of the rod, r the crank radius and s the amplitude of oscillation at the small end, then the acceleration of the piston is proportional to $r\omega^2$, the Coriolis acceleration to $2\omega\nu s$ and the relative acceleration to $\nu^2 s$. The usual oscillation frequencies will not exceed the angular velocity by a factor greater than 10, and the oscillation amplitude at the small end will be less than $r/500$. The relative acceleration is then 1/5 and the Coriolis acceleration 1/25 of the piston acceleration.

At high rpm figures the forces due to the gas pressure on the piston become small in comparison with the inertia forces. This permits easier consideration of these forces by assuming a coincidence of the extreme values in the actual and the theoretical indicator diagram. The intermediate values may then be neglected in this treatment.

DAMPING FORCES

Problems of dynamic instability arise from the periodic nature of the acting forces. While instability generally only accurs at speeds far higher than are employed in practice, damping forces must nevertheless be considered, as otherwise a completely false picture of dynamic instability will result, even in the range of speeds normally encountered.

Unfortunately it is difficult to estimate the effective forces due to damping, as this is composed of damping by fuel-compression, air-damping, and damping in the two bearings, which is of a hydrodynamic nature. All these obey different laws, and a damping force proportional to speed must be assumed.

STRESSES

In formulating the equations of motion, Hamilton's Principle may be utilised, as this will yield not only the differential equations but also the appropriate dynamic boundary conditions. Approximate solutions must be employed; the longitudinal and transverse oscillations are treated independently, the former appearing as disturbance terms in considering the latter. It will appear that the inertia forces due to the relative movement of the rod have no effect on the longitudinal oscillations, as can easily be seen.

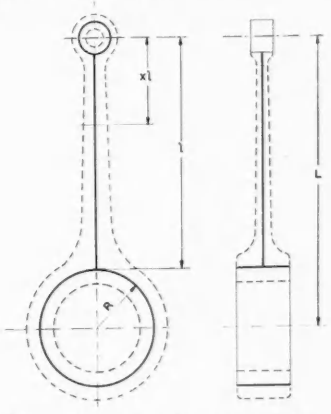


Fig. 1. **Schematic representation of connecting rod.**

After some calculation, the following equations are obtained for the nominal stresses (for explanation of the symbols used, see below):—

$$\text{Longitudinal stress } \sigma_l = -\rho\,\omega^2 l^2 (p_0 + p_1 x) \quad .. \quad (1)$$

$$\text{Bending stress } \quad \sigma_b = \pm \frac{E}{l^2} y\, r\, \alpha^2 \Psi \sin \alpha x \quad .. \quad (2)$$

The negative sign indicates compression. x is a dimensionless co-ordinate in the direction of the rod axis (see Fig. 1).

The following formulae are valid for the quantities p_0 and p_1, which depend on the crank-angle φ:

$$p_0 = -\frac{M_k r}{\rho F l^2}(\cos\varphi + \lambda \cos 2\varphi) + \frac{p_{\max} F_k}{\rho\,\omega^2 F l^2} \sum_{n=0}^{4} p_{jn} \cos \frac{n\varphi}{j} \quad .. \quad (3)$$

$$p_1 = -\frac{\lambda}{\mu}(\cos\varphi + \lambda \cos 2\varphi) \quad .. \quad (4)$$

The quantity Ψ, also a function of φ, has the form of a Fourier series,

$$\Psi = \sum_{m=0}^{\infty} g_m \cos \frac{m\varphi}{j} + \sum_{m=1}^{\infty} h_m \sin \frac{m\varphi}{j} \quad (5)$$

the coefficients g_m and h_m of which are given by the following system of recurrence formulae:

$$\left.\begin{aligned}
&2\beta_0 g_0 + \beta_1 g_1 + \beta_2 g_2 + \beta_3 g_3 + \beta_4 g_4 = 0\\
&-\frac{2m^2}{j^2} g_m + \sum_{n=0}^{4} \beta_n (g_{m-n} + g_{m+n} + g_{-m+n} + \frac{2m}{j} A h_m = 0\\
&-\frac{2m^2}{j^2} h_m + \sum_{n=0}^{4} \beta_n (h_{m-n} + h_{m+n} - h_{-m+n})\\
&-\frac{2m}{j} A g_m = \begin{cases} 2\Phi_k, \text{ when } m/j = k \\ 0, \text{ when } m/j \neq k \end{cases}\\
&(k = 1, 2 \ldots) \qquad (m = 1, 2, 3 \ldots)
\end{aligned}\right\} \quad (6)$$

Quantities with negative indices must be understood to be equal to 0. The coefficients $\beta_0, \beta_1 \ldots \beta_4$ are given by the following indentity:

$$\beta_0 + \beta_1 \cos\frac{\varphi}{j} + \beta_2 \cos\frac{2\varphi}{j} + \beta_3 \cos\frac{3\varphi}{j} + \beta_4 \cos\frac{4\varphi}{j} = \frac{\alpha}{2\alpha - \sin 2\alpha + 4\alpha E_2 \sin^2\alpha}[-\alpha(2\alpha - \sin 2\alpha) p_0 - (\alpha^2 - \alpha \sin 2\alpha - \sin^2\alpha) p_1 + \alpha^3 (2\alpha - \sin 2\alpha) E_0 + \alpha^2 (\alpha^2 - \alpha \sin 2\alpha - 3\sin^2\alpha) E_0 a + 4\alpha^2 E_4 \sin^2\alpha] \quad (7)$$

and the constants Φ_1 and Φ_2 follow from

$$\Phi_1 \sin\varphi + \Phi_2 \sin 2\varphi = \frac{4}{2\alpha - \sin 2\alpha + 4\alpha E_2 \sin^2\alpha}\left[\frac{\sin\alpha - \alpha\cos\alpha}{\alpha}\mu \sin\varphi + \frac{1-\cos\alpha}{2}\lambda \sin 2\varphi\right] \quad .. \quad .. \quad (8)$$

In equations (6) to (8), the following dimensionless abbreviations are used:

$$A = \frac{(2\alpha - \sin 2\alpha) E_5 + 4\alpha E_3 \sin^2\alpha}{2\alpha - \sin 2\alpha + 4\alpha E_2 \sin^2\alpha} \quad .. \quad (9)$$

$$\left.\begin{aligned}
E_0 &= \frac{E\mathcal{J}_0}{\rho\,\omega^2 F l^4} & E_1 &= \frac{E\mathcal{J}_1 R}{k l^2}\\
E_2 &= \frac{\Theta}{F R^2 l} & E_3 &= \frac{K}{\rho\,\omega F R^2 l}\\
E_4 &= \frac{\mu}{1-\mu}\frac{\mathcal{J}_1}{\mathcal{J}_0} E_0 & E_5 &= \frac{\vartheta}{\pi}\frac{\omega_b}{\omega}
\end{aligned}\right\} \quad (10)$$

List of Symbols Employed in Equations (1) *to* (10):

L ..... length of connecting rod between centres
l ..... length of column
$\lambda = r/L$
$\mu = l/L$ . . . length ratio
y ..... distance of neutral axis from surface
$\varphi = \omega t$ . . . crank angle, taken as positive in the direction of rotation, starting from top dead centre at the beginning of expansion
ω_b .... flexural natural frequency of the rod, approximately $\omega_b = \frac{\pi^2}{L^2}\sqrt{\frac{E\mathcal{J}}{\rho F}}$ using mean values of $\mathcal{J}$ and F
ρ ..... density of material
F ..... mean cross-section of rod
$\mathcal{J}$ ..... moment of inertia of rod (for bending in the plane of the crank)
$\mathcal{J}_0$ ..... moment of inertia at $x = 0$, i.e., approximately at the transition to the small end, disregarding the radius at the neck
$\mathcal{J}_1$ ..... moment of inertia at $x = 1$, i.e., at the transition to the big end, neck-radius being disregarded
$a = \frac{\mathcal{J}_1}{\mathcal{J}_0} - 1$ . . . auxiliary quantity
F_k ..... area of piston
M_k .... mass of piston
$p_{\max}$ .... max. pressure (ignition pressure)
K ..... coefficient of viscous damping in big-end bearing: $K = \frac{d^4\pi}{4h}\eta$
d ..... diameter of crank pin
h ..... half the bearing-play in big end
η ..... dynamic viscosity of lubricating oil at the running temperature
k ..... experimental stiffness factor at the transition from column to big end
E ..... modulus of elasticity of rod material
δ ..... logarithmic decrement of the free, undamped flexural oscillation of the rod = 0·001 to 0·05
α ..... smallest root of the equation $\frac{\tan\alpha}{\alpha} = \frac{1-\mu}{\mu}\frac{1}{E_1\alpha^2 - 1}$
j ..... index figure: $j = 1$ for two-stroke engine, $j = 2$ for four-stroke engine
Θ ..... geometrical moment of inertia of big end, about crank-pin axis
p_{jn} .... coefficients of eq. (3), see Table I

TABLE I.

n	0	1	2	3	4
p_{1n}	0·288	0·425	0·175	0·075	0·037
p_{2n}	0·151	0·283	0·302	0·183	0·081

From the calculated nominal stresses, fluctuating between maximum and minimum values, the actual extreme values of the stress can now be obtained by multiplication with the corresponding stress-concentration factors.

NUMERICAL EXAMPLE

The connecting-rod chosen as an example is shown in Fig. 2. The maximum speed is about 2200 rpm.

The units employed in the calculation are cm, kg, sec. The relevant data are: $L = 29$ cm, $l = 23$ cm, $R = L - l = 6$ cm, crank-radius $r = 7$ cm. Mean cross-section of rod $F = 3{\cdot}24$ cm². Moment of inertia of the column at the small-end $\mathcal{J}_0 = 4{\cdot}9$ cm⁴. Moment of inertia of column at big-end (radii neglected) $\mathcal{J}_1 = 8{\cdot}43$ cm⁴. Moment of inertia of big end, with bearing and bolts $\Theta = 3980$ cm⁴. Ignition pressure $p_{max} = 55$ kg/cm². Piston area $F_k = 95$ cm². Weight of piston with pin and rings $G_k = 2{\cdot}302$ kg. Index figure $j = 2$ (four-stroke). Ratio $\lambda = 0{\cdot}241$. Length ratio $\mu = 0{\cdot}793$.

Using a modulus of elasticity $E = 2{\cdot}1 \times 10^6$ kg/cm², and a specific mass $\rho = 8 \times 10^{-6}$ kg cm⁻⁴ sec², and introducing the quantity $\xi = 100/\omega$, where ω is the angular velocity of the crankshaft, we get $a = 0{\cdot}72$, and from eq. (10) $E_0 = 142\xi^2$, $E_2 = 1{\cdot}48$, $E_4 = 938\ \xi^2$.

To allow for the considerable radius at the column to big-end transition, k is taken as infinity, and thus $E_1 = 0$.

For the calculation of viscous damping in the big-end bearing, a dynamic viscosity $\eta = 0{\cdot}002 \times 10^{-4}$ kg cm² sec is assumed. The diameter d of the crankpin is 7 cm, and h is taken as 0·005 cm. Then $K = 0{\cdot}075$ kg cm sec, and $E_3 = 0{\cdot}035\ \xi$.

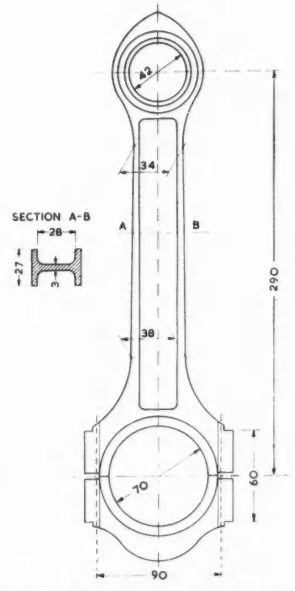


Fig. 2. **Drawing of connecting rod treated in numerical example.** (Dimensions in mm.)

In calculating the oscillation-damping of the shaft, ϑ is taken as 0·005. The angular velocity ω_b of the flexural oscillation is estimated (no accurate value is necessary here) as 8600 sec⁻¹. This yields $E_5 = 0{\cdot}137\ \xi$.

From equations (3) and (4) and using Table I, we have

$$p_0 = 5{\cdot}75\ \xi^2 + 10{\cdot}8\ \xi^2 \cos\frac{\varphi}{2} + (11{\cdot}5\ \xi^2 - 1{\cdot}38)\cos\varphi + 7\ \xi^2 \cos\frac{3\varphi}{2} + (3{\cdot}1\ \xi^2 - 0{\cdot}332)\cos 2\varphi$$

$$p_1 = -0{\cdot}304\cos\varphi - 0{\cdot}073\cos 2\varphi$$

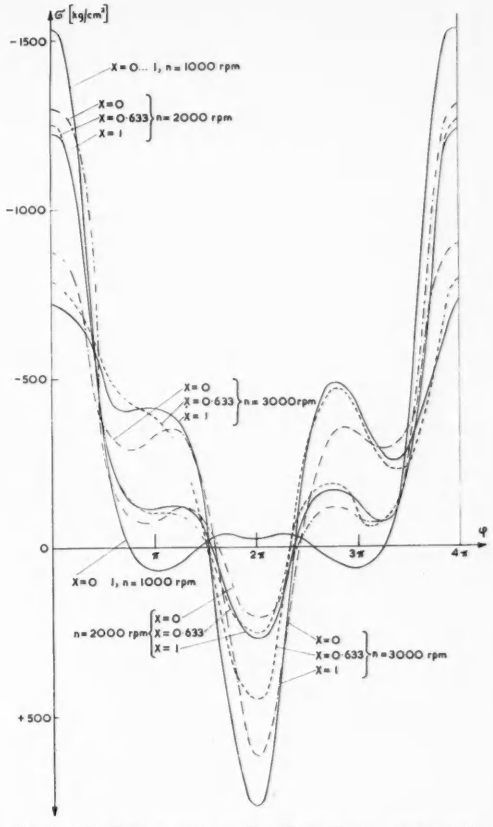


Fig. 3. **Nominal stresses as functions of the crank angle at various speeds.**

With $E_1 = 0$, α becomes $= 2{\cdot}553$. Eqs. (7) to (9) then yield

$A = 0{\cdot}088\ \xi$, $\beta_0 = 6510\ \xi^2$, $\beta_1 = -39{\cdot}7\ \xi^2$, $\beta_2 = 5{\cdot}7 - 42{\cdot}3\ \xi$, $\beta_3 = -25{\cdot}6\ \xi^2$, $\beta_4 = 1{\cdot}35 - 11{\cdot}4\ \xi^2$, $\Phi_1 = -0{\cdot}312$, $\Phi_2 = -0{\cdot}083$.

To determine the stresses at various speeds, the set of equations (6) must now be solved. In the expansion (5) we must go at least as far as $m = 4$.

Equations (6) will then have the form

$$2\beta_0 g_0 + \beta_1 g_1 + \beta_2 g_2 + \beta_3 g_3 + \beta_4 g_4 = 0$$

$$2\beta_1 g_0 + \left(-\frac{1}{2} + 2\beta_0 + \beta_2\right) g_1 + (\beta_1 + \beta_3) g_2 + (\beta_2 + \beta_4) g_3 + \beta_3 g_4 + A h_1 = 0$$

$$-A g_1 + \left(-\frac{1}{2} + 2\beta_0 - \beta_2\right) h_1 + (\beta_1 - \beta_3) h_2 + (\beta_2 - \beta_4) h_3 + \beta_3 h_4 = 0$$

$$2\beta_2 g_0 + (\beta_1 + \beta_3) g_1 + (-2 + 2\beta_0 + \beta_4) g_2 + \beta_1 g_3 + \beta_2 g_4 + 2A h_2 = 0$$

$$-2A g_2 + (\beta_1 - \beta_3) h_1 + (-2 + 2\beta_0 - \beta_4) h_2 + \beta_1 h_3 + \beta_2 h_4 = 2\Phi_1$$

$$2\beta_3 g_0 + (\beta_2 + \beta_4) g_1 + \beta_1 g_2 + \left(-\frac{9}{2} + 2\beta_0\right) g_3 + \beta_1 g_4 + 3A h_3 = 0$$

$$-3\,A\,g_3 + (\beta_2 - \beta_1)\,h_1 + \beta_1\,h_2 + \left(-\frac{9}{2} + 2\,\beta_0\right)h_3 + \beta_1\,h_1 = 0$$

$$2\,\beta_4\,g_0 + \beta_3\,g_1 + \beta_3\,g_2 + \beta_1\,g_3 + \beta_1\,g_3 + (-8 + 2\,\beta_0)\,g_4 + 4\,A\,h_4 = 0$$

$$-4\,A\,g_4 + \beta_3\,h_1 + \beta_2\,h_2 + \beta_1\,h_3 + (-8 + 2\,\beta_0)\,h_4 = 2\,\Phi_2$$

These can be solved by simple iteration, beginning with h_4, as the terms containing β_0 are very large in comparison with the others. In the present case, exact values are obtained with the first step. We immediately obtain (in order of solution)

$$h_4 = -\frac{1{\cdot}28}{\xi^2} \times 10^{-5} \qquad h_2 = -\frac{4{\cdot}8}{\xi^2} \times 10^{-5}$$

$$h_3 = -\frac{0{\cdot}02}{\xi^2} \times 10^{-5} \qquad h_1 = -\frac{0{\cdot}01}{\xi} \times 10^{-5}$$

$$g_4 = +\frac{0{\cdot}35}{\xi^3} \times 10^{-9} \qquad g_2 = +\frac{0{\cdot}65}{\xi^3} \times 10^{-9}$$

Later terms are smaller by another order of magnitude. It can be seen that only the coefficients h_2 and h_4 need be retained. These give the transverse oscillations which are in phase with the load. The other terms become significant only at very high speeds.

The longitudinal stress σ_l and the flexural stress σ_b are then given by eqs. (1) and (2)

$$\sigma_l = \frac{E}{l}\,u_x = -\rho\,\omega^2\,l^2\,(p_0 + p_1\,x) =$$

$$= -\left\{244 + 456\cos\frac{\varphi}{2} + \left[487 - \frac{1}{\xi^2}(58{\cdot}4 + 12{\cdot}9\,x)\right]\cos\varphi + 296\cos\frac{3\varphi}{2} + \left[131 - \frac{1}{\xi^2}(14{\cdot}1 + 3{\cdot}1\,x)\right]\cos 2\varphi\right\}$$

$$\sigma_b = \pm\frac{E}{l^2}\,y\,r\,\alpha^2\,\Psi\sin\alpha\,x = \pm\,3{\cdot}25 \times 10^5\,\Psi\sin\alpha x$$

$$= \pm\,[(15{\cdot}6\sin\varphi + 4{\cdot}16\sin 2\varphi)\sin\alpha x]/\xi^2$$

A mean value of $y = 1{\cdot}8$ cm is used in the above.

Fig. 3 shows the resultant stress $\sigma = \sigma_l + \sigma_b$ as a function of φ for various speeds. The stresses were calculated at the sections $x = 0$, $x = 0{\cdot}633$ ($\sin\alpha\,x = 1$) and $x = 1$.

While the inertia forces exceed the gas-expansion forces at high speeds, they appear negligible at 1000 rpm, and thus there is no difference in stress between the sections $x = 0$ and $x = 1$.

In the column, no stress concentrations occur, and thus the nominal stresses may be taken as the actual ones. At the transition to the small-end, however, a stress-concentration factor of 1·45 is assumed* by which the nominal values must be multiplied.

A comparison with fatigue tests on the material employed can now be made, taking into account the effects of surface finish.

*See Mickel-Sommer-Wiegand: Berechnung und Gestaltung der Triebwerke schnellaufender Kolbenkraftmaschinen. Berlin, Springer Verlag, 1942.

SWITZERLAND

Simplified Outdoor-Type High-Speed Air-Blast Circuit Breakers with Voltage Ratings up to 380kV

By H. THOMMEN. (From *The Brown Boveri Review*, Vol. 37, Nos. 4/5, April-May, 1950, pp. 123-136, 22 illustrations.)

The need for a considerably simplified form of air-blast circuit breaker has resulted in a construction without a built-on isolating switch in series with the main contacts. This article gives details of the design and operation of the new circuit breaker, which incorporates multiple breaks for higher voltage ratings.

OIL as the classical arc-extinguishing medium now has a strong rival in compressed air. The oil-free air-blast circuit breaker has undisputed advantages in important indoor installations, and extensive practical experience in recent years has confirmed that the air-blast circuit breaker is the ideal switching device, even for the highest operating voltages for which outdoor switchgear is generally provided.

OUTDOOR CIRCUIT BREAKER DESIGN AND OPERATION

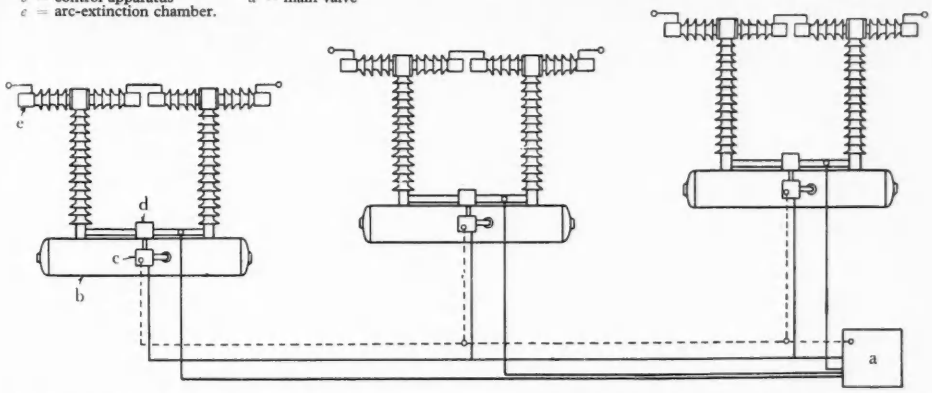
The new circuit breaker has no built-on isolating switch in series with the arcing contacts. On switching out, the opening of the main valves causes the arc-extinction chambers to be filled with compressed air (Fig. 1). After the arc has been subjected to the air blast, the exhaust valves on the extinguishing chambers close, while the main valves remain open, so that insulation between the contacts is established by the compressed air. Switching-in is again achieved by means of the main valves. As these are of the three-way type, the compressed air is discharged from the extinguishing chambers at the same time as the valves close, and the contacts are free to close. The system is thus extremely simple, without any mechanical connecting rods passing from earth to parts under voltage, and ensures maximum operational reliability.

The arc-extinction chambers have been the subject of particularly detailed studies. When the compressed air enters, the arcing contacts are moved into the arc-extinguishing position and are held there for several half-cycles by means of a pneumatic device which is independent of temperature, the arc being extinguished by the compressed air. The switch movement ends and the discharge of compressed air through the exhaust valve is stopped when an equalising space, provided in the arc-extinction chamber, is also filled with compressed air. The air flow in the system is such that the circuit breaker can handle the heaviest short-circuits with practically no wear at the sealing surfaces. Normal arc durations amount to 1 or 2 half-cycles.

For the closing process, pistons are provided in addition to springs. The pistons are actuated by compressed air, stored in an intermediate receiver, as soon as the pressure in the arc-extinction chamber decreases at a sufficiently rapid rate. Closing occurs at a residual pressure of several atmospheres, and a closing

Fig. 1 (below). **Outline drawing of the simplified high-speed air-blast circuit breaker.**

——— = Pneumatic pipes - - - - - = Electrical connections
a = Apparatus cabinet *b* = air receiver
c = control apparatus *d* = main valve
e = arc-extinction chamber.



arc is thus practically avoided. Oscillograms of opening and closing operations are shown in Figs. 2 and 3.

BEHAVIOUR UNDER RUPTURING STRESSES

Adaptability due to multiple breaks. The multiple breaks with potential control used make possible a simple circuit breaker construction suitable for severe operating requirements and high service voltages. In Fig. 4, the lowest curve shows the characteristic of a single break. Improved contact design has made it possible to keep the breaking capacity approximately constant over a wide range of voltages. This property is also retained for higher voltage values by employing several breaks. With 10 breaks, for example, the circuit breaker remains insensitive to voltage rises even above the 380-kV limit, and this is of importance since a modern circuit breaker is required not only to clear normal short-circuits, but also to open with two systems or parts of a system 180 deg. out of phase.

The advantages of multiple breaks are only fully utilised when the voltage distribution across the open breaks is as uniform as possible. The effect of capacitance to earth is reduced by connecting capacitors or resistors to the extinguishing chambers, for voltage compensation. The simplest design includes control capacitors arranged on the arc-extinction chambers which are most highly stressed. For higher power ratings, additional resistances are connected across all the chambers.

Fig. 5 shows the equalising effect of outer control capacitors as determined by measurements made on a 220-kV circuit breaker. A practically ideal distribution is achieved with interconnected systems 180 deg. out of phase, so that satisfactory interruption under out-of-step conditions is also ensured with partial capacitive control.

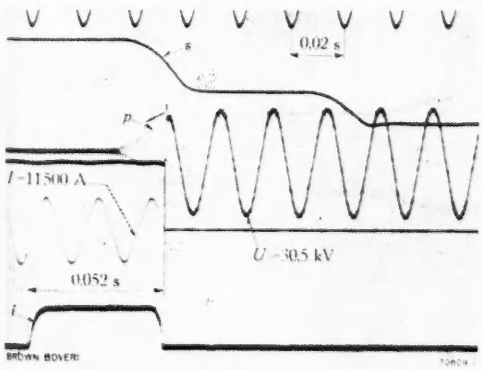


Fig. 2. **Oscillogram of interruption with one arc-extinction chamber.**

s = Contact movement ; *p* = pressure in the chamber ; *I* = breaking current (r.m.s. value of a.c. component at instant of contact separation) ; *U* = recovery voltage (r.m.s. value) ; *i* = current in trip coil.

Note the two-stage movement of the arc-extinction contacts and the rapid current interruption.

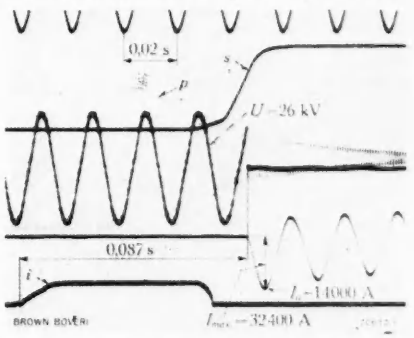


Fig. 3. **Oscillogram of closing operation with one break on short-circuit.**

s = Contact movement ; *p* = pressure in the chamber ; *U* = voltage prior to short circuit (r.m.s. value) ; I_a = short-circuit current (r.m.s. value of a.c. component) ; I_{max} = maximum peak value of current ; *i* = current in closing coil.

At contact making there is still compressed air in the chamber so that arcing is practically eliminated.

The non-linear Resorbit resistors used for higher power ratings have a steep voltage-dependent characteristic. At the beginning of a voltage rise, the amount of current absorbed is still small. As the disturbing capacitances on the circuit breaker are small, a few milliamperes are, however, sufficient to obtain a satisfactory voltage distribution. The ideal distribution is thus already reached at a fraction of the voltage at which the race between the re-striking voltage and the de-ionisation of the break reaches a decisive stage. The small residual resistance current is interrupted by auxiliary contacts after the voltage peaks occurring during the switching operation have disappeared. In a typical application, a 220-kV circuit breaker with partial control for 3500 MVA symmetrical breaking capacity, the capacity can be increased to 5000 MVA by means of built-on resistances which are located in the vertical and auxiliary contacts in the inclined insulators.

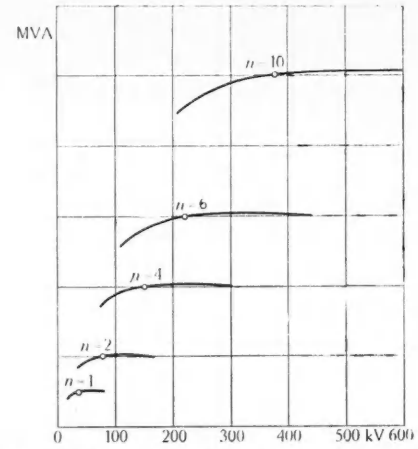


Fig. 4. **Breaking capacity in relation to voltage with different numbers of breaks.**
The circles represent the normal service points of the circuit breakers which withstand voltage rises. These are, therefore, particularly suitable as tie breakers.

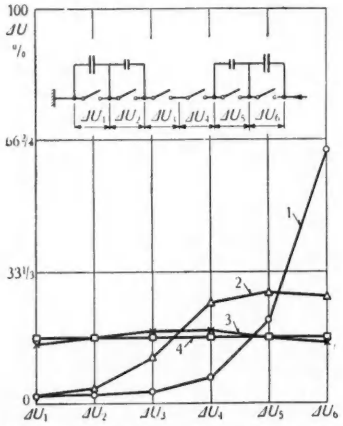


Fig. 5. **Voltage distribution across breaks of a 220-kV high-speed air-blast circuit breaker with six arc-extinction chambers.**
1 = Without potential control; one breaker terminal earthed;
2 = with partial capacitive control (one breaker terminal earthed);
3 = with partial capacitive control (systems 180 deg. out of phase);
4 = with resistance control.

Breaking capacity tests. The behaviour of the entire circuit breaker can be assessed when the potential distribution and rupturing properties of an individual break are known. The test can be made with only a fraction of the power normally required. The break rapidly loses its conductivity after the passage of the current through zero, so that no detrimental effects due to residual currents have to be taken into account.

The air-blast circuit breaker, with its symmetrical compressed-air supply to the individual breaks and its uniform switching process, shows far less scatter in test results than the oil circuit breaker with its complicated vaporisation and decomposition processes occurring in the oil. For this reason, it has been possible to evolve a simple breaking-capacity formula. Comparative tests have been carried out on individual breaks with a power as near as possible to their full breaking capacity, and with a 150 kV high-speed air-blast circuit breaker, in which the flow diameter of the nozzle contacts was reduced to enable the limiting capacity of the modified system to be determined with certainty with the short-circuit power available in the test plant. Comparisons with the breaking capacity of the individual breaks have proved the correctness of the assumptions made, this being true for both types of potential control. With the resistances, the expected proportional increase in breaking capacity with the number of breaks has been confirmed (Fig. 6). In conjunction with the results previously obtained, this is further proof of the fact that the individual testing of the breaks is founded on a physically sound basis, whereas this is not the case with the inevitably incorrect extrapolation method used for extra-high-voltage circuit breakers having a small number of breaks, regardless of the type of circuit breaker.

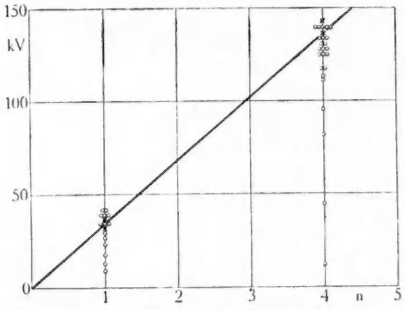


Fig. 6. **Increase in voltage and breaking capacity with increasing number of breaks.**
The tests were carried out with one and four breaks and constant breaking current. To test a circuit breaker with four breaks, tests with one-fourth of the full breaking capacity on one single arc-extinction chamber are sufficient.

Interrupting reliability and natural frequency conditions. Much has been written in recent years regarding natural frequency effects. The highest frequency values are the decisive factor, but it should be noted that in extensive networks the maximum power never coincides with the highest natural frequency. It is thus necessary to consider the system conditions in order to avoid erroneous conclusions calling for circuit breakers of unnecessarily large size. The experimental results available have shown that the rate of de-ionisation of the break is always greater than that of the rise in restriking voltage across the circuit breaker terminals. In 220-kV systems, for instance, the natural frequencies

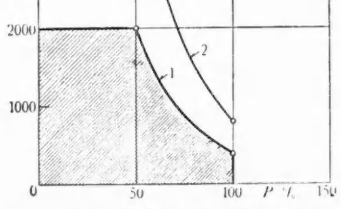


Fig. 7. **Interrupting reliability of a 220-kV circuit breaker under natural frequency conditions in relation to breaking capacity.**
Breaking capacity is given as a percentage of the guaranteed capacity.
Curve 1 = Proposal for new Swiss circuit breaker specifications;
Curve 2 = values actually attained with type of circuit breaker described.

which occur fall within the shaded area of Fig. 7. The solid line 1 bounding this area forms the basis of the new Swiss standard specifications for circuit breakers. The lower stresses occurring in service thus provide an additional safety factor which is not present in the case of breakers insensitive to frequency effects, since these operate with equal difficulty under all natural-frequency conditions.

Interruption of low and medium currents. In addition to the clearance of short-circuits, the interruption of long unloaded lines and low inductive currents has also been the subject of detailed studies. When switching operations are undertaken on long unloaded lines, re-striking of the arc across the break has to be avoided, and the oscillogram of Fig. 8 demonstrates the excellent behaviour of a 220-kV circuit breaker used for this purpose on a 474 km long overhead line.

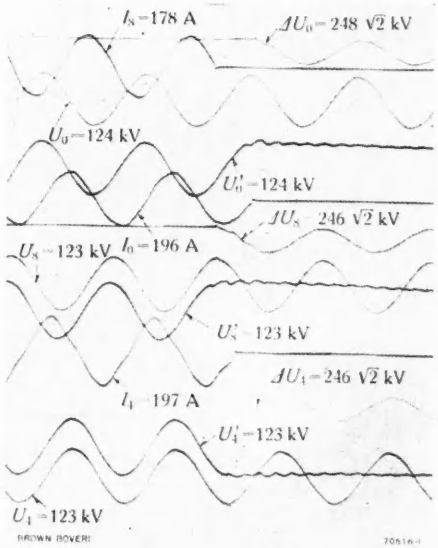

I_8 = 178 A
ΔU_0 = 248 $\sqrt{2}$ kV
U_0 = 124 kV
U_0' = 124 kV
ΔU_8 246 $\sqrt{2}$ kV
U_8 = 123 kV
I_0 = 196 A
U_8' = 123 kV
ΔU_4 = 246 $\sqrt{2}$ kV
I_4 = 197 A
U_4' = 123 kV
U_4 = 123 kV
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Fig. 8. **Disconnection of a 474-km overhead line with a 220-kV high-speed air-blast circuit breaker under no-load conditions. No restrikes occurred.**
U_0 U_4 U_8 = Phase voltages on the busbar side (r.m.s. values);
U_0' U_4' U_8' = phase voltages on line side (r.m.s. values);
ΔU_0 ΔU_4 ΔU_8 = voltages across circuit breaker units (peak values);
I_0 I_4 I_8 = phase currents (r.m.s. values).

In the case of inductive currents, the overvoltages occurring depend not only on the circuit breaker, but also on the circuit characteristics. Unloaded transformers are easier to disconnect than large compensating reactors, owing to the favourable effect of the iron losses. More difficult conditions are sometimes encountered with arc-extinction coils, but the non-linear auxiliary resistors with their large current absorption capacity can in such cases produce considerable improvement, whereas they are not required for transformers unless these are of a very large size.

INSULATION OF THE ARC-EXTINCTION CHAMBERS

Compressed-air insulation. After a breaking operation has been completed, the arc-extinction chambers may remain under voltage at both ends and can, therefore, be subjected to particularly high stresses. A characteristic feature of the new high-speed air-blast circuit breakers is that the arc-extinction chambers are subjected to an internal air pressure in the open-contact position. The question of insulation by means of compressed air has been very carefully investigated, with a view to increasing the internal insulation strength in the compressed-air chamber in the fully-open position to such an extent that it is considerably higher than the external insulation in free air.

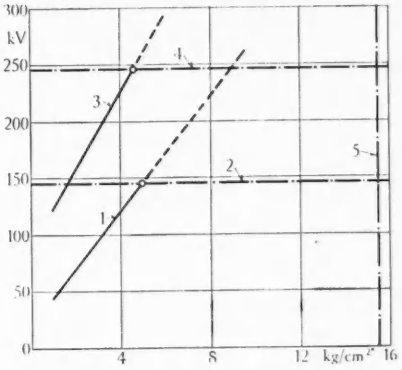


Fig. 9. **Electric strength of an arc-extinction chamber in relation to the pressure.**
1 = Internal electric strength at 50 cps (kV r.m.s.);
2 = external electric strength at 50 cps (kV r.m.s.);
3 = minimum internal impulse strength (kV peak);
4 = minimum external impulse strength (kV peak);
5 = rated pressure of breaker.

The insulation strength between simple electrodes increases rapidly with rising pressure. In switchgear, with its complicated field distribution, success depends to a great extent on the correct construction and dimensions of the internal parts. Fig. 9 shows the results obtained with the final design of arc-extinction chamber which has been evolved, and illustrates the high insulation strength achieved with both impulse stresses and tests at normal frequency over a range extending well below the rated pressure of the circuit breaker. Although no condensation should occur if the compressed-air installation is properly arranged, for safety reasons tests were made with eroded contacts having a roughened surface and with the inside surface of the porcelain in a very wet condition, and these tests gave similar high insulation values. Thus, deterioration of the insulation is out of the question, regardless of the number of interruptions, whereas this is not the case with oil circuit breakers in which the oil is subject to ageing and becomes carbonised.

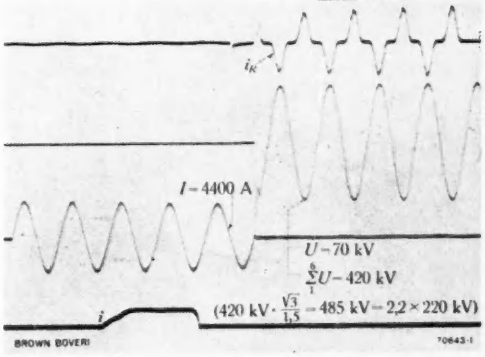


Fig. 10. **Oscillogram of an opening operation with increased voltage stressing of arc-extinction chambers.**
U = Recovery voltage per arc-extinction chamber (r.m.s. value);
I = breaking current (r.m.s. value of a.c. component at instant of contact separation); i_R = current in Resorbit resistor (the characteristic curve is a result of the non-linear resistor); i = current in trip coil.

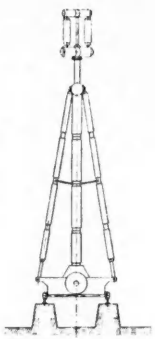
External insulation. A thorough investigation was made of the external insulation of several extinction chambers arranged in series. In the case of a large number of breaks, by skilful arrangement of the additional capacitors, flash-over voltages are attained which satisfy standard specifications and ensure reliable operation under the most extreme stress conditions occurring in service. Tests were carried out with an increased voltage applied both with the circuit breaker in the off position for a long period and at the instant of opening. Typical results are shown in Fig. 10 for the case of a clearance of a short-circuit, with a re-striking voltage of 70 kV per switch chamber, corresponding to a re-striking voltage of 420 kV (with a line voltage of 485 kV) across a 220 kV breaker. Very good results were also obtained with the air-blast circuit breaker during wet tests, owing to the favourable effect of the horizontally located arc-extinction chambers.

MANUFACTURE, INSTALLATION AND OPERATION

The wide range of sizes of circuit breakers is always based on the same fundamental elements, such as arc-extinction chambers, main valves, etc. Designs for higher voltages are obtained by increasing the number of arc-suppression chambers and lengthening the supporting insulators.

The three interchangeable single-phase units of a circuit breaker are connected together solely by compressed-air pipes and control cables, without rods or coupling shafts. Simultaneous operation of the three breaker units is best achieved by connecting the closing and opening coils in parallel. The switch position is indicated by a pneumatically-operated signal. In case of a breakdown in the control circuit for one breaker, a pneumatic safety control system ensures the switching of all the units to the "off" position.

For extra-high-voltage systems it is preferable, for economic reasons, solidly to earth the transformer neutral points. In this case, single-phase high-speed re-closing is also of great importance in order to be able to clear the single-phase-to-earth arcing short-circuit, a frequent cause of disturbance, without service interruption. The circuit breakers can be used at any time with a suitable combination of protective relays, without additional elements, for single or three-phase high-speed re-closing. Off-on-off switching operations are easily dealt with. A short break time of three cycles is achievable, and to obtain the shortest possible dead time, adjustment can be made without considering the circuit breaker.



Back-signalling, emergency and manually operated pneumatic control systems are located in an apparatus cabinet, and on the circuit breaker a control unit contains all the devices for preventing pumping and incompleted switching operations ; this has proved its reliability in indoor and outdoor installations.

Experience over a period of more than ten years has shown that no dangerous condensation occurs in the arc-extinction chambers, which are filled with air under pressure, provided the compressed-air installation is correctly set up. The air receivers of the breaker are designed so that two complete switching cycles can be carried out with the compressed air contained in them, and the breaker thus remains ready for operation even if there is a temporary drop in pressure in the air distribution system. The small cavities in the insulators are continuously ventilated with dry air and the air consumption thus entailed is negligible.

As the isolating switch in series with the main contacts has been dispensed with, the problem set by the moving masses is simplified correspondingly and the breakers operate absolutely free from vibration.

EXTENSION OF RATED VOLTAGE TO 380 kV

Sweden is the first country to have introduced a 380-kV transmission system on a wide basis, in order to connect the power stations in the north of the country to the load centres in the south. The inherent conditions of the Swedish 220-kV system already imposed severe demands on the circuit breakers, with line voltages rising to 280 kV under certain conditions. Moreover, arc-suppression coils are employed so that the single-phase breaker units are subjected to heavy stresses when clearing simple earth faults. Comparative tests with different types of circuit breakers have shown that air-blast circuit breakers with multiple breaks are the most suitable for such conditions.

In the 380-kV system, the short-circuit currents do not generally exceed those usually encountered in conventional extra-high-voltage systems, but the voltage differences over long transmission distances cause extra stresses on the circuit breaker, owing to charge equalisation after interruption of current. Still higher voltages may result from large capacitive loads occurring temporarily when compensating reactors or active loads are suddenly switched off. Such unfavourable conditions may, however, be avoided by correct planning and operation. To allow an ample margin for service conditions, the 380-kV breakers for Sweden are designed to isolate a 500 km line at a voltage of 600 kV, this corresponding to a capacitive load of over 500 MVA.

The tests have been carried out in a special high-voltage laboratory, and no extrapolated values are used. Special screens have been developed for the insulation to earth to obtain the requisite dielectric strength, in conjunction with the potential control devices. Design details of these breakers are given in Fig. 11. Additional bracing insulators are provided to take the force of the wind. The breakers are also able to isolate satisfactorily compensating reactors up to 180 Mvar, while ensuring high-speed re-closing.

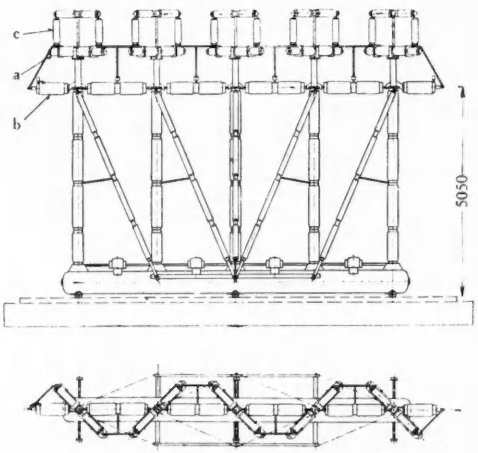


Fig. 11. **380-kV high-speed air-blast circuit breaker with a current rating of 1000 A and a breaking capacity of 8500 MVA.**

Breaker equipped for high-speed re-closing. *a* — Arc-extinction chamber ; *b* — control capacitors ; *c* — resistance.

The Possibilities of Combined Gas and Steam Turbine Installations

By P. CHAMBADAL. (From *La Technique Moderne*, Vol. 42, Nos. 17-18, September 1-15, 1950, pp. 277-285, 5 illustrations.)

In recent years, various studies have been made of the possibility of suitably combining gas turbines and steam turbines. These have in some cases been somewhat prematurely interpreted as an admission of failure of the straight industrial gas turbine, whereas in fact the purpose of the studies has been to endeavour to utilise the advantages that can be derived from a new type of heat machine. A wide variety of combinations has been considered, and these will be briefly described in the following lines.

PURPOSE OF THE VARIOUS COMBINATIONS CONSIDERED

1. A reduction in boiler size, weight and cost may be obtained by increasing the air supply to the combustion chamber by means of a compressor, driven by a gas turbine in which the combustion gases expand to atmospheric pressure. This makes it possible to increase considerably the heat transmission coefficients and to reduce the size of the combustion chamber and the area of the heating surfaces. This principle is used in " Velox ", " Sural ", and various other installations. By raising the pressure of the air to that of the steam generated, the " equal pressure boiler cycle " developed by E. Mercier[1] is obtained. In boilers working on this principle, the pressures inside and outside the boiler tubes are equal, so that thinner tube walls and smaller overall boiler dimensions can be used.

2. Instead of increasing the power/weight ratio of the steam installation, one can try to increase this ratio for the gas turbine installation by, for instance, connecting the air compressor to a steam turbine so as to obtain at the alternator terminals, after allowing for inevitable losses, all the power developed by the gas turbine shaft[2].

3. More generally, by suitable combinations of various components it may be possible to raise the power/weight ratio of the entire system and to develop a combination which is particularly suitable for special applications, such as marine propulsion[3]. A special case is that of a combined installation erected in an American power station[4], where the exhaust gases from the gas turbine have been used to preheat the water supplied to the boilers, thus saving an equivalent amount of steam and increasing the power of the existing steam turbines without resorting to boilers of a greater capacity.

4. Finally, a combination of the two means of energy production may also serve to improve the overall thermal efficiency. It is this thermodynamic aspect of the problem that will be dealt with in detail in the following sections. However, even this problem may be regarded from two different points of view:—

(a) The installation is solely developed for electric power generation. The power rating is then already determined and the object is to find the saving in heat (expressed in heat units, and not considering fuel costs) obtainable by employing a combination of gas and steam turbines in place of an installation using only either one or the other type of machine.

(b) Besides generating electricity, the installation is also designed to supply a given quantity of heat. This heat is then provided for a definite purpose and can only be transported over a limited distance, so that the heat supply per hour is well determined in each case. The electrical machines of the installation can, however, be connected to the grid so that any possible excess in energy produced is always utilised. In such a case, the addition of a gas turbine to the steam installation may then result in an increase in the electrical energy produced without altering the heat production. The problem is then to compare the cost, in heat units, of this additional electrical energy to that required for obtaining it by means of either condensation turbines or gas turbines alone. The two cases (a) and (b) will therefore have to be considered separately.

THERMAL EFFICIENCY OF CONDENSING TURBINES AND GAS TURBINES

It will be assumed that the steam turbine, in the same way as the gas turbine, is used only for the production of electrical energy and, therefore, exhausts into a condenser.

The thermal efficiency of the best gas turbines developed is of the order of 33-34 per cent and does not, therefore, differ very much from that of the most economical steam turbine installations. The first question is whether this equivalence is purely fortuitous or whether a plausible explanation can be found.

Any improvements to a heat machine tend to make its real cycle more similar to an ideal cycle regarded as a theoretical limit. For the steam turbine, the ideal cycle is the Carnot cycle with two isothermals and two reversible adiabatics, whereas for the gas turbine it is the Ericsson cycle with two isothermals and two constant-pressure curves. The Carnot and Ericsson cycles are, however, equivalent if they are described within the same extreme temperature limits. Thus, the thermal efficiencies of the steam turbine and the gas turbine tend towards the same theoretical value.

In the case of the gas turbine, the realisation of the Ericsson cycle is only prevented by the deficiencies of the means used, but not by the motive fluid; in the steam turbine, the realisation of the Carnot cycle or an equivalent cycle is only conceivable if the steam remains always in the saturated state. Under this condition, the reversible heating of the water condensate to the boiling temperature T_2 by means of a continuous withdrawal of steam has the effect of substituting in place of the Rankine cycle *IBCDI* of Fig. 1 the cycle *IBCJI*, where the curve *CJ* is parallel to *BI*. This latter cycle is, however, equivalent to the Carnot cycle *ABCDA* described between the same temperature limits T_2 and T_1.

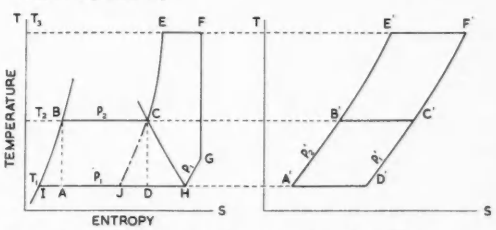


Fig. 1. **Comparison of the thermal cycles of the steam turbine and gas turbine.**

Under idealised conditions, in the case of the gas turbine it is conceivable that a continuous reheating of the motive fluid during the expansion process *B'C'* in Fig. 1, and continuous cooling during the compression *D'A'*, in conjunction with a complete recovery of the heat yielded by the exhaust gases during their cooling process *C'D'*, will provide a cycle in which the reheating of the fluid between the temperatures T_1 and

T_2 over the length $A'B'$ will not require any expenditure of fuel. The resultant cycle $A'B'C'D'A'$ is an Ericsson cycle which is also equivalent to the Carnot cycle $ABCDA$.

The isothermals T_1 and T_2 of cycles $IBCJI$ and $A'B'C'D'A'$ correspond, however, to transformations which are physically different. In the steam cycle, along these isothermals the fluid receives or gives off heat without producing or absorbing mechanical energy. Apart from the work absorbed by the feed pump, mechanical energy appears in the cycle only along the pseudo-adiabatic CJ, the shape of which expresses the gradual loss of weight of the steam passing through the turbine. In the gas cycle, the isothermal transformations correspond to pressure variations for the expansion $B'C'$ or compression $D'A'$ of the motive fluid, and purely thermodynamic processes occur only along the constant-pressure lines p_1' $(C'D')$ and p_2' $(A'B')$.

The conditions are no longer equivalent when the steam cycle operates with superheated steam. By superheating and continuously resuperheating the steam along EF under optimum conditions, it is found that the expansion line FG is situated entirely within the superheated region (for simplicity, Fig. 1 does not take account of the changes in weight of the steam during expansion). The use of part of this steam for the superheating of the condensate makes this operation non-reversible, so that the elimination of the " triangular loss " $AIBA$ is no longer as complete as in the previous case. Moreover, the upper part of the cycle now includes two isothermals separated by a constant-pressure line, viz., BC and EF, the latter being physically equivalent to $E'F'$, instead of a single isothermal. In order to reduce the resultant loss in efficiency, a constant-pressure line GH is then introduced in the lower part of the cycle. The physical properties of steam do not make it possible, therefore, to achieve a Carnot-type cycle limited by the superheating temperature and the temperature of the cold source. This inferiority of the steam turbine is further accentuated by the fact that the permissible inlet temperature is lower than that which can be adopted for the intake of the gas turbine.

However, the inevitable shortcomings of the equipment used have a much greater effect on the thermal efficiency of the gas turbine than on that of the steam turbine. In the case of the latter, the compression work (i.e., the work of the feed pump) is relatively unimportant, so that the efficiency of the expansion is the main factor affecting the thermal efficiency of the cycle. With the gas turbine, the efficiency of the compressor must also be included, and this reduces considerably the useful work obtained for a given expenditure of heat.

Finally, it is a fact that the efficiencies of the real steam turbine and gas turbine cycles are also equivalent, provided the heat-recovery efficiency of the gas turbine system is given a sufficiently high value. The experience so far available is insufficient to determine whether the choice of a high heat-exchanger efficiency is economically justified or which is the most economical efficiency for this component. For the time being, it will therefore be considered that the two cycles have substantially equal efficiencies, and our problem will be to try to determine the extent to which a combination of these two types of machines is capable of producing an overall efficiency superior to that obtained by either of the constituent units.

EFFICIENCY OF SIMULTANEOUS PRODUCTION OF ELECTRICAL ENERGY AND HEAT

Let W = electric power (kWh) developed at the generator terminals, and Q = quantity of heat (kcal/h) supplied to the heating system. If the installation comprises several generators (for instance, if, besides having a back-pressure or auxiliary steam turbine, it includes a gas turbine driving an independent generator), W will denote the total power output of the installation. Furthermore, let H = quantity of heat (kcal/h) supplied by the fuel. Then, whether the steam turbine is operating alone or in a combination with a gas turbine, the thermodynamic efficiency of the installation is usually given by

$$\eta = \frac{860\ W + Q}{H} \qquad .. \quad (1)$$

where 860 kcal/h/kWh is the thermal equivalent of the kilowatt-hour.

This expression is widely used and seems at first sight to be fully justified, but it is not, however, particularly suitable for the purpose for which it is intended, that is, for the assessment of the merits of an installation from the point of view of the heat expended. It has, in fact, the peculiar property of conducing to lower efficiencies for higher values of W, so that with this definition the simultaneous production of electrical energy and heat would always be less economical than the production of heat alone.

This can be illustrated by considering a simple example, viz., the substitution of a back-pressure turbine in place of an installation for the production of steam for heating purposes only. For the simple heating installation, eq. (1) becomes

$$\eta_b = Q/H \qquad .. \quad (1')$$

where η_b = boiler efficiency ($\eta_b < 1$). The difference $H - Q > 0$ takes account of the losses of the installation. Let it be assumed that the boiler efficiency remains unchanged when, instead of producing steam at the pressure required for heating, steam is supplied at a higher pressure, it being also utilised for useful work by expanding it in a back-pressure turbine. In this case, the additional term 860 W is afflicted with relatively higher losses than the term Q, since it will take account not only of the losses resulting from steam production but also of overall bearing friction losses, steam leakage losses, increased power absorbed by auxiliaries, such as the feed pump, and in some cases the electrical losses of the generator. Thus, if we denote by H_e the additional heat expenditure required for the production of electrical or mechanical energy, and by η_e the corresponding efficiency, we have

$$\eta_e = 860\ W/H_e < \eta_b$$

and, therefore, $\eta < \eta_b$, η being the efficiency of the combined system.

The difference $\eta_b - \eta$ increases proportionally to the energy produced by the back-pressure turbine, although this energy only requires an expenditure of 1200 kcal/kWh, whereas even in a modern large-size power station the production of this energy without simultaneous supply of heat requires an expenditure at least twice as great.

This anomaly is due to the fact that eq. (1) has been established by regarding as useful output all the energy which has not been lost, with an abstract interpretation of the fact that this energy is present in two physically different forms. This type of abstraction is justified if it is desired to make a heat balance of a process, but is not admissible when seeking to determine its efficiency. For the heat balance, one verifies the relationship: energy used + losses = energy expended. In efficiency considerations, the conversion of various types of energy into heat units requires further examination. The calorie of work and the calorie of heat are dimensionally equal but not true equivalents. To illustrate this, it may be pointed out that the production of a work calorie requires an expenditure of at least three heat calories, whereas, conversely, a work calorie may not only be completely converted into heat (for instance, by Joule effects) but may also be used to drive a heat

pump and thus produce a thermal equivalent of several calories.

Further evidence of the inadequate assessments obtained with expressions such as eq. (1), which take no account of the physical nature of each of the useful types of work produced, is given by the following comparison:— An efficiency of 60 per cent may be regarded as extremely high when a large portion of the work is produced in the form of electrical or mechanical energy, whereas the same value of 60 per cent is regarded as most uneconomical when the useful work consists almost exclusively of energy used for steam heating.

These considerations apply equally well in the case of a steam turbine working in combination with a gas turbine. The expression for efficiency given by eq. (1) thus requires modification, so that η will really be a thermal efficiency for the installation, and not a quantity varying inversely with the power developed at the generator terminals. The modification which is now proposed is simple enough: it consists in introducing an " equivalence factor " k in eq. (1), in order to reduce all the useful effects to the same form of energy, either work calories or heat calories.

It is thus proposed that the overall thermal efficiency be defined by

$$\eta = \frac{860\,W + k\,Q}{H} \qquad .. \qquad (2)$$

where $k < 1$. This relationship now enables us to determine directly the effect of an increase in the term with W (obtained, for instance, by the addition of a gas turbine) for a given value of Q. Writing both sides in differential form, we have

$$d\,\eta = \frac{860\,d\,W - \eta\,d\,H}{H}$$

or

$$d\,\eta/\eta = 860\,d\,W/(\eta\,H) - d\,H/H$$

so that

$$\frac{d\,\eta}{\eta} = \frac{d\,W}{W + \dfrac{k\,Q}{860}} - \frac{d\,H}{H} \qquad .. \qquad (3)$$

Therefore, the variation in efficiency will be positive if

$$\frac{d\,W}{W + \dfrac{k\,Q}{860}} > \frac{d\,H}{H}$$

that is, if the relative increase of the useful work is greater than that of the energy required.

VALUE OF THE EQUIVALENCE FACTOR k

The value of the equivalence factor k used in eq. (2) is not a universal constant like Joule's mechanical equivalent of heat, but depends on the technical and economical conditions prevailing throughout the time period which is being considered. For given conditions, the value of k will be obtained by comparing the quantity of heat Q with the quantity of electrical energy which it would be possible to produce with the same expenditure of fuel in an installation with either a condensing turbine or a gas turbine used for this purpose only. The fuel required for heating purposes only is determined from eq. (1') as

$$H = Q/\eta_b \qquad \text{[kcal/h]}$$

and if η_e denotes the thermal efficiency for electrical energy production, the electric power generated will be

$$W = H\,\eta_e/860 \qquad \text{[kWh]}$$

Then, in accordance with the above definition, k is obtained from the relationship

$$k\,Q = 860\,W$$

or

$$k\,H\,\eta_b = H\,\eta_e$$

so that

$$k = \eta_e/\eta_b \qquad .. \qquad .. \qquad .. \qquad (4)$$

Thus, for instance, if $\eta_e = 0{\cdot}28$ and $\eta_b = 0{\cdot}80$, then $k = 0{\cdot}35$. The production of a quantity of heat Q for heating purposes should, therefore, only be taken into account in eq. (2) after reducing its value by 65 per cent in the present case.

DETERMINATION OF MAXIMUM ECONOMY

There are two different aspects in economy considerations. One of these is how to obtain the highest possible efficiency η for a combined installation in which the heat energy and the electric power output can be given the values required for this purpose. This is not the object of the present study. The other is the determination of conditions for obtaining a minimum value for the amount of fuel H used for the entire installation, when the quantity of heat Q to be produced is already fixed by local requirements.

Considering again the general case of a simultaneous production of work and heat, let Q = heat transmitted to the steam for heating purposes, and W = power supplied to the network by a generator driven by a back-pressure turbine not associated with a gas turbine. The hourly fuel consumption of such an installation is given by eq. (2):

$$H = [860\,W + k\,Q]/\eta \qquad \text{[kcal/h]}$$

On the basis of the foregoing assumptions, the combination of the steam turbine with a gas turbine will leave unchanged the values of Q and k, but will raise the total power output to $W' > W$ and correspond to a thermal efficiency η' which may be different from η. The new fuel consumption is then

$$H' = [860\,W' + k\,Q]/\eta'$$

and is generally greater than H.

The additional power $W' - W$ produced with the gas turbine avoids the use of a condensing turbine producing the same power and having a fuel consumption

$$H'' = [860\,(W' - W)]/\eta''$$

where η'' = thermal efficiency of this second steam turbine. The economy obtained with the gas turbine is, therefore,

$$E = H'' - (H' - H) \text{ [kcal/h]} \qquad .. \qquad (5)$$

This economy E may be positive even if $\eta' < \eta$; it may therefore be an advantage to reduce the thermal efficiency of the installation if this reduction makes it possible to increase considerably the power developed under sufficiently economical conditions.

If various combinations of gas and steam turbines are to be compared, the combination with the greatest advantages will be that which leads to a maximum value for E, or, since H is constant, to the maximum of the difference ΔH

$$\Delta H = H'' - H' = (860\,W''/\eta'') - H' \qquad (6)$$

It is evident that if no heat energy is produced ($Q = 0$), the corresponding fuel consumptions are $H = 860\,W/\eta$ and $H' = 860\,W''/\eta'$, and assuming $W' = W$, the economy is $E = 860\,W'\,[(1/\eta) - (1/\eta')]$, which is an obvious result. Furthermore, since W''/η'' is then constant, the value of the difference in fuel consumption is $\Delta H = -860\,W''/\eta'$. This expresses the fact that the maximum economy corresponds to the maximum efficiency of the combined installation.

For preliminary calculations of the comparative efficiencies of various steam and gas turbine combinations, it is useful to reduce the number of variables to a minimum. For the gas turbine installation, this is achieved by adopting constant values for the gas intake temperatures, the efficiencies of the components and the pressure head losses through the various

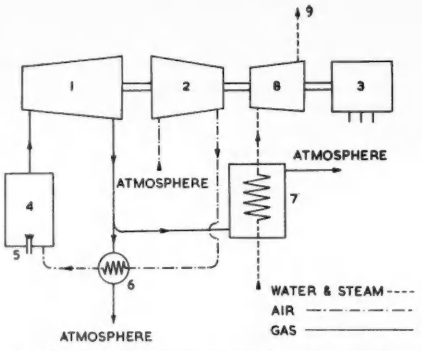


Fig. 2. **Schematic diagram of combined gas turbine and back-pressure steam turbine installation.**

(1) Gas turbine; (2) compressor; (3) generator; (4) combustion chamber; (5) fuel burner; (6) heat exchanger; (7) boiler; (8) steam turbine; (9) steam for heating purposes.

elements of the circuit. For the steam turbine (of the condensing or back-pressure type), the calculation is simplified by adopting an invariable expansion line (and hence a well-defined final state) for the steam. Thus, the number of independent variables is reduced to two, viz., the maximum inlet pressures of the two fluids. In certain cases, the arrangement of the connections between the gas and steam installations limits the number of possibilities for the maximum pressures of the two fluids; in such cases, only compatible values need be considered.

For steam installations, it is usual to assume that the quantity of heat carried away by 1 kg of steam at the turbine exit has a constant value, since, in a back-pressure turbine, the final condition of the steam is determined by the operating conditions of the heating equipment and, in a condensing turbine, it is determined by the temperature of the circulating water and the permissible moisture content of the turbine exhaust. Similarly, in the case of the gas turbine, it may be assumed that (1) the enthalpy or total heat of the gases at the turbine exit has a constant value, and (2) the addition of the gas turbine has no effect on the operating conditions of the steam turbine, that is, on the state of the steam at the turbine inlet, the possible reheating of the condensate, etc.

In a back-pressure or condensing turbine, the expansion work of one kg of steam and the heat required for the generation of this steam both remain constant, so that the efficiency of the steam installation is the same whether this installation is combined with a gas turbine or not. However, in a gas installation, each kg of gas enters in the form of atmospheric air at ambient temperature and comes out as combustion products at a constant temperature, according to the assumptions made above. However, whereas in a steam installation operating independently, combustion takes place at atmospheric pressure so that the gas is neither compressed nor expanded and, therefore, performs no work, in a combined installation the expansion work of the gas is greater than the work required for its compression, and this excess is added to the expansion work of the steam, thus increasing the useful amount of work produced. The superposition of the two thermal cycles thus results in an increase in useful work which entails only a slight increase in fuel consumption.

The above conclusions are based on a constant ratio for the weights of gas and steam. If the type of combination chosen requires an increase of this ratio, the loss in sensible heat of the gases per kg of steam also increases. Further losses occur if the introduction of the gas turbine brings about a rise in temperature of the exhaust gases, or if the initial pressure and temperature conditions of the steam are reduced to lower values, even without altering the final state of the steam.

EXAMPLES OF COMBINED INSTALLATIONS

1. *Boiler heated by the exhaust gases of the gas turbine.* The water is evaporated and superheated by the heat of the exhaust gases after expansion in the gas turbine (see Fig. 2). Fuel is used only in the combustion chamber (4) of the gas turbine. The exhaust gases pass through the boiler (7) before being discharged. The steam expands in the steam turbine (8) which, for the sake of simplicity, is shown on the same shaft as the gas turbine (1), the air compressor (2) and the electric generator (3). The exhaust (9) of the steam turbine is connected to a condenser or to heating equipment. In the latter case (back-pressure turbine conditions), the flexibility of the installation is increased by providing a heat-exchanger (6) in parallel with the boiler (7). When less steam is required for heating, the exhaust gases give off their heat to compressed air, which makes it possible to maintain the production of electrical energy at a high level with adequate efficiency.

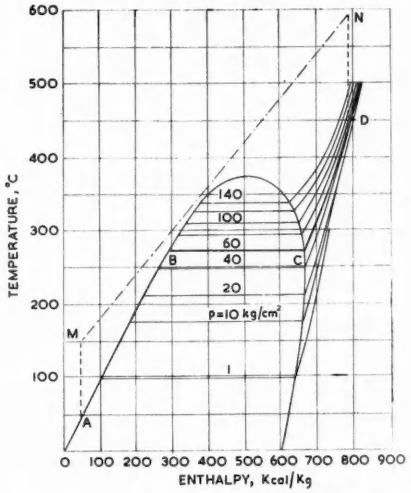


Fig. 3. **Temperature-enthalpy diagram for steam.**

The operating characteristics of this installation are limited by the requirement that positive temperature differences must exist between the fluids transmitting and receiving heat at both ends of the boiler and throughout the transformation of the water from the liquid state to that of superheated steam. Expressed by means of an enthalpy diagram (see Fig. 3), this means that the broken line *ABCD* thus obtained must be situated below the straight line *MN*, which represents, in the same diagram, the variation of a weight of gas corresponding to 1 kg of steam. It is, therefore, apparent that, in order to reduce the gas temperature at the boiler exit, a very high boiler inlet temperature is required. This can be achieved by providing the gas turbine with several combustion chambers. In the extreme case, one of these combustion chambers would be situated between the gas turbine and the boiler, so that the gas turbine would then be supplying only part of the heat required for steam generation.

A study of Fig. 3 shows that this difficulty is more easily overcome if the temperature and pressure of the steam are reduced. Moderate pressures and temperatures are, however, much more suitable for back-pressure turbines where the steam consumption is

usually relatively low than for condensing turbines which are generally designed for greater steam consumptions.

2. *Utilisation of the exhaust gases of the turbine as an additional fuel in the combustion chamber of the boiler.* There are two possible arrangements. The first consists of a gas turbine with only one combustion chamber, which supplies its exhaust gases to the boiler combustion chamber, where they mix with the fuel injected. A test installation of this type has been developed by the Oerlikon Company, and has shown the advantages of recovering the heat of the gases coming out of the turbine.

The second arrangement is to supply the gas turbine with compressed air which has passed through a surface-heater placed in the path of the combustion gases coming from the boiler. This arrangement and the one previously mentioned are both due to M. W. Karrer[5,6]. The second arrangement is shown in Fig. 4. The heat produced in the boiler (4) is used for steam generation, heating of the compressed air (7), and the reheating of the feedwater (10). The installation includes an air turbine (1), an air compressor (2), and a heat exchanger (6). Although this system requires a surface heater unit, it has the advantage of being able to operate on solid fuels and of supplying turbine (1) with pure air only.

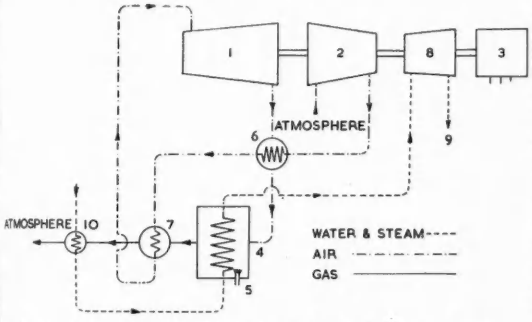


Fig. 4. **Schematic diagram of a combined air turbine and back-pressure steam turbine installation, proposed by M. W. Karrer.**

(1) Air turbine; (2) compressor; (3) generator; (4) boiler; (5) fuel burner; (6) heat exchanger; (7) air heater; (8) steam turbine; (9) steam for heating purposes; (10) feedwater heater.

In both systems, the air mixed with the fuel cannot be reheated, so that the temperature of the boiler combustion gases can only be reduced by reheating the feedwater. This reheating is economical in the case of a back-pressure turbine, where it makes it possible to avoid the use of a low-pressure unit for the reheating of the water. In the case of a condensing turbine, however, reheating the water by means of the combustion gases would reduce the amount of bled steam and, therefore, decrease the thermal efficiency of the steam cycle.

3. *Supercharging the boiler by the gas turbine compressor.* If, instead of arranging the boiler combustion chamber so that it will receive the air which has expanded through the gas turbine, this combustion chamber is placed ahead of the turbine, the cycle shown in Fig. 5 is obtained. The boiler (4) is supercharged by the compressor (2), and the combustion gases expand in the turbine (1), which develops power greater than that required by the compressor. This excess power is added to the power produced by the steam turbine (8). A combustion chamber (7) is in parallel with the boiler (4); and in parallel with the heat exchanger (6) is a feed heater (10) which is supplied with part of the exhaust gases from the turbine (1). There are thus two circuits for the compressed air

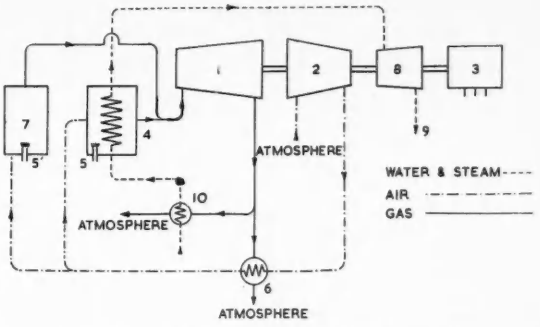


Fig. 5. **Schematic diagram of a combined gas turbine and back-pressure steam turbine installation and pressure-fired boiler.**

(1) Gas turbine; (2) compressor; (3) generator; (4) boiler; (5) (5') fuel burners; (6) heat exchanger; (7) combustion chamber; (8) steam turbine; (9) steam for heating purposes; (10) feedwater heater.

coming from compressor (2), and two circuits for the exhaust gases of turbine (1). The corresponding rates of flow are adjustable according to the requirements for steam heating and electric power. A variation in the amount of steam required corresponds to a variation in the same direction of (*a*) the compressed air output going into the boiler (4) and (*b*) the exhaust gases passing through the feed heater (10). Similarly, a variation of the electric power required is obtained by varying in the same direction the air and gas output quantities supplied to the combustion chamber (7) and the heat exchanger (6).

This combination has interesting properties when its operating characteristics are chosen so that an appreciable decrease may be obtained in the temperature of the gases in the feedwater heater. However, the same conclusion applies in this instance as in the previous case: feed-heating by means of the exhaust gases from the turbine is an advantage only if it is not used in place of feed-heating by bled steam. The proposed arrangement therefore gives greater economy in the case of a gas turbine combined with a back-pressure turbine than when the steam turbine is provided with a condenser and only serves for electrical energy production.

CONCLUSIONS

Published data and calculations made by the author show that the gain obtained by combining a gas turbine with a condensing turbine varies between 0·01 and 10·0 per cent, according to operating conditions. This gain relates only to the fuel consumption, so that the combination may be justified for other reasons than those previously considered. For instance, the arrangement of Fig. 5 enables a considerable reduction in the boiler size as a result of pressure-firing the boiler with the gas-turbine driven compressor. Likewise, this conclusion is not applicable to the "equal pressure cycle" mentioned at the beginning of this article, which includes novel units such as free-piston compressors. These components may have very high individual efficiencies which may result in an important increase in the overall efficiency of the system[1].

In combination with the gas turbine, the back-pressure turbine cycle gives far more favourable results. It has also a further advantage not yet mentioned: for a given value of the power supplied by the steam, the steam consumption is evidently much more important in the back-pressure cycle than in the condensing cycle. This has the result that the gas consumption and gas power are also greater in the first case than in

the second. In practice, this additional power is of the same order as the power developed by the back-pressure turbine operating alone, and the expenditure corresponding to this additional power is in some cases only of the order of one half of that of an equivalent condensing turbine[7]. Thus it may be said that the combination of the back-pressure turbine with a gas turbine may result in an economy of the order of 30-50 per cent in the consumption of the additional stage delivering an extra amount of power equal to that of the original stage operating independently.

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U. S. A.

Evaluating Tool Performance

By T. BADGER. (From *The Tool Engineer*, Vol. 25, No. 3, September, 1950, pp. 17-20, 3 illustrations.)

Selection of a cutting tool material is complicated by the fact that each of the three major classes—high-speed steel, cast alloy, and tungsten carbide—is available in several types and grades, and all of these are offered by many manufacturers.

IN order to evaluate cutting tool materials, it is necessary continually to conduct tests to formulate comparative results. A method of tool evaluation has been developed in the Headquarters Manufacturing Engineering Laboratory, which accurately predicts shop results for cutting tool materials. This test provides a check on the efficiency of stock tools in a few hours. It is a simple lathe test, and, although it may seem that such a test is not an adequate standard for a tool material that may be used in dozens of different applications, experience has shown that the test will accurately predict whether a tool material will perform satisfactorily or fail when used in the shop.

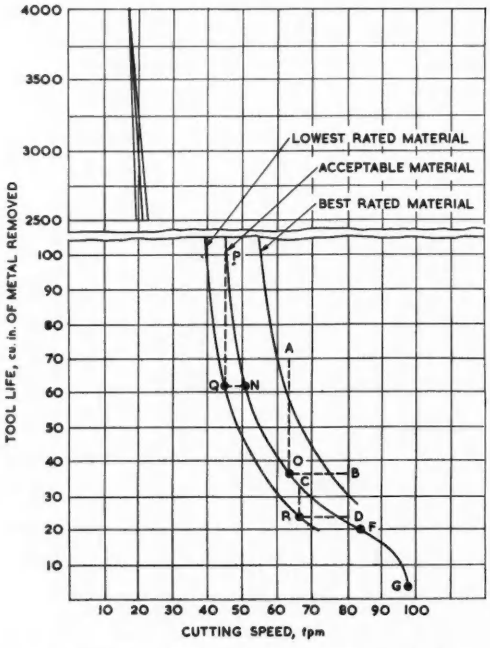


Fig. 1. **Tool life plotted against cutting speed for three tool materials.**

A lathe-type test gives good results because no matter how complex a cutting tool may be, when reduced to essentials, any single point on a complex cutting tool acts exactly like a tool when used as a lathe tool.

Fig. 1 shows three speed-tool life curves. First we will consider only the curve labelled as an "acceptable material." Note that for all values of cutting speed lower than the point F, the curve is a smooth parabola, so that as cutting speed is decreased, tool life increases. It is obvious from the shape of the curve that tool life increases much faster than the decrease in cutting speed. Note also that for cutting speeds higher than point F, the curve simply drops off and tool life rapidly approaches zero as cutting speed increases. This indicates the cutting speed at which tool failure will be almost instantaneous.

It might appear that the determination of this point would provide a good criterion of a tool material's efficiency. However, laboratory experience has shown that tests based on determining this point are not reliable. In fact, this area beyond F of any speed-tool life curve is very erratic, and data taken on this portion of the curve are of no practical value. Points beyond this knee of the curve must be avoided in cutting tests. As an economy measure, high test speeds are desirable, but it has been found that a cutting speed about 15 to 20 per cent below the cutting speed at the knee of the curve will produce accurate results with the highest economy consistent with good results. In setting up any new test, the first step is to make sufficient preliminary tests to determine the position of the knee on the speed-tool life curve.

SPEED-TOOL LIFE RELATIONSHIP

The three curves on Fig. 1 are all of the same shape and are proportionally equidistant from each other throughout their length on the cutting speed axis. This is true throughout the range of speeds normally used in production, but at very slow speeds the curves will tend to come together. This is of no importance, as the object of all tool development is to determine the highest practical cutting speed, rather than the lowest cutting speed. A change in cutting feed, depth of cut, machinability, tool design, coolant, or severity of

the cut will shift the position (but not the general shape) of the speed-tool life curve.

Fig. 2 is a typical curve that illustrates the effect on tool life as cutting feed is varied (other factors remaining fixed). Note that this curve is suggestive of a cotangent curve, but over the greater portion of its length—which fortunately is for the usual range of commonly used shop feeds—it is very close to a straight line. For our purposes, it is necessary to know only that for constant cutting speed the speed-tool life curve will be displaced up or down at the slope of this curve. For the conditions shown in Fig. 2, a feed of from 0·020 to 0·030 in. would be best, and the feed at the point of intersection of the actual and assumed curves would probably be the best value.

The depth of cut will also affect the speed-tool life curve. For depths of cut less than the nose radius of the tool, a variation in depth of cut has a great effect on tool life; but after the depth of cut reaches twice the radius, the effect of depth of cut is slight. All that is necessary is to select a depth of cut deep enough to be in the area where small errors in depth setting will not cause large errors in tool life values. At the same time, a low depth of cut is desired for economy. For all practical purposes, a depth of cut of $\frac{1}{8}$ in. has been found to give satisfactory test results.

TEST PRACTICE

In test practice, test logs are made from a single lot of steel and heat-treated at the same time in quantities sufficient to last for several years. This is to eliminate variations in machinability of the test material. However, small variations between logs and on different sections of the same log exist. A test coupon is taken from each log and any logs showing a variation of more than ± 10 Brinell are rejected. This variation is minimized by testing as many tools as possible in sequence. This tends to average out any unavoidable variations in machinability. Incidentally, when using cast iron logs, it has not been found possible to get test logs cast to the hardness tolerances shown above, so it has been necessary to accept logs within a range of ± 20 Brinell. In the case of cast iron, it is known that microstructure has a far greater effect on machinability than hardness, and a microphotograph is therefore made of each cast iron test log. Logs showing wide variation in structure and graphite distribution are rejected. When changing from one lot of test logs to another, a machinability test is made to determine the relative machinability of the two lots, and test speeds are adjusted to suit the new lot of test material.

STANDARDIZED TOOL DESIGN

Tool design and the quality of the tool grind will also affect the results of tool tests. The effects of tool design are eliminated from the test results by using standardized tools. Such tools may vary in size, but the shape remains constant and is controlled by cam grinding to ensure that the tools are uniform. All tools are diamond-lapped as a final operation. Each tool is Rockwell tested to ensure uniform hardness. Efforts have been made to control the surface finish of the tool grind by profilometer readings. This method was rejected in favour of a standardized grinding procedure and visual inspection prior to use.

All routine tool testing is done without a coolant. Coolant testing is carried out as another activity, and is not included in tool testing, as this only introduces the possibility of another variable to influence test results.

The severity of the cut has a profound influence on test results, particularly when testing carbide tool materials, as the severity of the cut must be matched to the grade of carbide being tested. For testing a tool under severe interrupted cuts, the test log is slotted with four slots so that the tool receives four severe

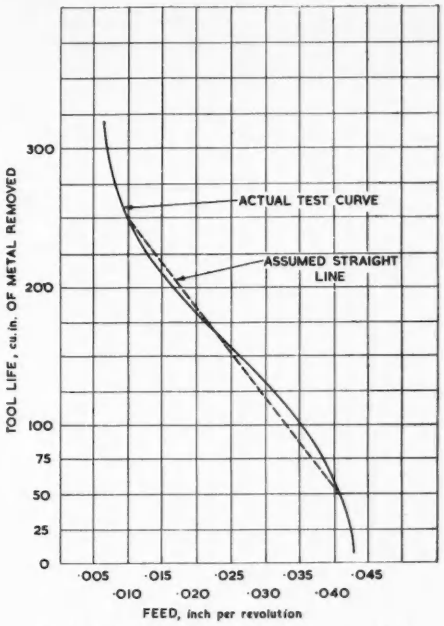


Fig. 2. **The effect of cutting feed on tool life.**

shocks per revolution. High-speed steel tools can be tested by normal procedure, but normal cast alloy and carbide tools will fail instantly under such a test. In this case a negative rake tool design is used. The design has a 20 deg. negative back rake with a 15 deg. positive side rake and a 0·010 in. flat ground on the cutting edge. It has been found that this test will distinguish tools as to quality for applications where shock is a major factor. The test is not used with other than carbide tools because it has been found that high-speed steel or cast alloy tools, under this type of test, will fall in the same general order as if the tools were subjected to the standard continuous cut test. This is far from true with carbides, and the test must be matched to the grade being tested. Here, considerable judgment is required, as the test must be matched to the average application of the carbide grade to be tested. For example, a grade of carbide for precision boring may be tested on continuous cutting at 1200 fpm, 0·010 in. depth of cut and 0·005 in. feed. At the other extreme, a grade of carbide for severe interrupted cutting may be tested on a slotted log at 130 fpm, $\frac{1}{4}$ in. depth of cut, and 0·022 in. feed, using a negative rake tool. Here too, care must be exercised to ensure that the various grades being tested are comparable.

One practical problem that immediately appears when any test is actually started is how to determine the end point for tool life. Several years ago, a tool was considered ready for regrinding only when actual tool failure occurred. It was later found that results were more accurate if the point where partial breakdown (minute chipping of the cutting edge at the nose radius) occurs, as indicated by a change in surface finish, was taken as a limit of tool life. This method still was not completely satisfactory, and therefore the point where a tool wore 0·002 in. (as evidenced by a change in work diameter of 0·004 in.) was used as the limit of tool life. This proved satisfactory, and was further useful because the results of tool tests should also establish data upon which to predict optimum cutting conditions.

Recently, a more accurate method has been developed, which consists of measuring the rate of tool wear by graphical methods from test results. Both of these methods have enabled test results to be duplicated with an accuracy of less than $\pm$ 2 per cent.

SETTING-UP TEST

In the foregoing discussion, each of the factors has been treated as if the other factors had no effect. This is far from true. Also, if a complete investigation of every factor were to be made for every test, the cost would be prohibitive. Actually, for any class of tool material and test material, the average cutting conditions are known from shop practice or previous test results. In a few instances, involving experimental materials, these facts are not known but can be approximated by actually turning a piece of the test material with the test tools. In setting up a tool test, the following general procedure is followed :—

1. A nominal value of feed (usually from 0·015 in. to 0·030 in.) and a nominal depth of cut (over 0·100 in.) is selected.

2. Using these conditions, a preliminary test is made to determine a cutting speed that will give the point of almost instant tool failure. This point is illustrated at G in Fig. 1.

3. Using this point as a basis, the previously described method is used to construct a speed-tool life curve such as GFON of Fig. 1. It is necessary only to make sufficient tests to determine the shape of the curve for cutting speeds about 30 per cent less than that where the knee of the curve occurs. In Fig. 1 the knee would be at F, where speed is 84 fpm. A cutting speed of about 80 per cent of this value, say 67 fpm, would then be selected for the standard test speed.

4. Using this speed and the previously selected depth of cut, only three feed tests need be made ; one at the original test feed, one about 0·005 in. higher, and one about 0·005 in. lower. If these tests are recorded on a graph such as Fig. 2 and the points fall on a reasonably straight line, the originally selected speed is satisfactory. If this is not so, sufficient tests must be made to determine the straight line portion of the feed curve and the preliminary speed test repeated, using the new value of cutting speed.

5. For any value of depth of cut in excess of 0·100 in. no adjustment is normally required, but if desired, check tests at a depth of a little greater and a little less than the selected value can be made to determine if small variations in depth of cut show excessive changes in tool life.

6. Using the above selected values, the test can be set up, the only precaution being that every effort should be made to keep all other test factors constant.

A test set up on this basis is one that can be relied upon to produce dependable results. But no matter how dependable or accurate the results, they are useless unless a proper interpretation of the results is made. The fundamental problem of a tool test is to compare tool materials one with another for the purpose of selecting the best. But the ratio of this difference can be misleading if not properly stated. The lower left-hand curve in Fig. 1 represents the average performance of the poorest tool material tested in a group of high speed steels. The middle curve represents a tool that has been selected arbitrarily as acceptable, that is, tools performing better than this standard are considered good tools. The right-hand curve represents the average performance of the best tool material of the group tested. Now, if we wished to make a comparison of the best tool material with the acceptable standard, we could do this at any point such as O on the curve of the standard material. If we wished to compare the resulting tool life of the test materials by using the percentage increase of tool life of the best material compared with the tool life of the acceptable standard as a measure of efficiency (along the line OA), the following would result : (60/36 $\times$ 100) — 100 = 62 per cent. If we wished to compare the resulting possible increase in cutting speed of the best material with the cutting speed of the acceptable standard as a measure of efficiency, the percentage increase in efficiency (along the line OB) would be : (75/63 $\times$ 100) — 100 = 19 per cent. In the same way, comparing the acceptable material with the lowest rated material at R and Q on the curve, the following is noted :

	Percentage decrease in cutting speed	Percentage decrease in tool life
at R	13	24
at Q	12	61

Note that, in general, the percentage change in tool life for a constant cutting speed has a greater absolute value than the percentage change in cutting speed for constant tool life. The above values do not prove the facts, but from test experience, the following is known :

(a) Over the major portion of the speed range, the two curves will have a constant value of the ratio of percentage change in cutting speed for constant tool life.
(b) The ratio of percentage change in tool life for constant cutting speed will be different at every point, and will increase as the cutting speed decreases.

It is obvious that the percentage change in cutting speed is a correct measure of a cutting tool's efficiency. However, sufficient area is required to make the desired comparison, and this is too expensive for routine testing. A method for routine tool testing is to run the test at a constant speed and to present the results in tool life expressed as cubic inches of metal removed. Note that although this exaggerates the actual percentage difference between materials, this exaggeration is much less for a test speed selected as previously described than one selected at random at lower values of cutting speed, nor does if affect the relative standing of the test tools. When more exact values for research purposes are desired, the test is continued long enough to make a complete comparison of the speed-tool life curves.

The final test value is taken as the common average of five tests made from tools of the same lot of tool material under test. The question naturally arises as to how closely a random tool of this material will come to the test value.

It is entirely possible—but not probable—that in a single comparison of material A with material B, the low value for A and the high value for B would result. Both of these values would be misleading. However, if a few more comparison runs are made, the average will perform close to the average predicted by the test. Another question that might arise is how close to the average will individual tools of a material perform? Obviously, tool material C may be better than tool material A, even if their averages were identical. Individual tools of material C may perform very close to the average value, but individual tools of material A may vary considerably. No satisfactory answer has been developed for this question. In practice, the material is rated in accordance with the average. This is correct because an answer is required to the question: " How will this tool perform on the average ? " Individual high or low values are soon averaged out in the shop.

The test procedure, as outlined above, has several advantages other than its use in tool evaluation. For example, the same procedure can be used for evaluating the effects of machinability, coolants, rake angles, tool design, quality of tool grind, speed, feed, and depth of cut. To make such tests as these, it is only necessary to allow the quality under consideration to vary in the test as the other factors are held constant. In addition, the test results can be used for the mathematical determination of cutting formulae.

CLASSIFIED ABSTRACTS

Numerous important abstracts which, for lack of space, cannot be published in our main editorial columns, are regularly included in condensed form in this section. Subscribers may obtain photostat copies of all original articles at cost.

ELECTRIC GENERATORS AND MOTORS

Theory of Three-Phase Shunt-Wound Commutator Motors

By V. KLIMA. (From *Electrotechnicky Obzor, Czechoslovakia*, Vol. 39, No. 13, 1950, pp. 260-280, 38 illustrations.)

THE paper presents, in two parts, the theory of stator-fed commutator motors. The first part contains a physical discussion based on simple mathematical relations and the Helmholz-Thévenin theorem.

The second part (theoretical appendix) gives the mathematical derivations. Exact physical equations with total reactance values (not leakage reactances) are used, assuming the usual conditions, i.e., the effects of saturation, iron losses and the variable brush contact resistance are neglected.

Equations for the secondary current I_2 are derived first by the method developed by Dr. Dreyfus, which procedure is mathematically simple, and the results can easily be discussed on a physical basis. It is shown that the commutator motor with a regulator can be considered as being fed by a current source on the secondary side, the voltage of which is the sum of all secondary no-load voltages, this voltage being applied to an equivalent impedance Z_2k, which is the sum of the impedances of the motor and all the regulators, when their primaries are short-circuited. This can also be shown directly by the Helmholz-Thévenin theorem.

A circle diagram for the general case of regulation and compensation is derived. Using elementary methods of conformal representation, a formula for the centre of the circle is obtained. The vector of the centre, expressed as the current I_{2s}, may be resolved into two components, one of which is constant and gives the centre of the circle for zero regulator voltage, while the other is proportional to the no-load voltage of the regulators. This second vector may thus (after change of scale) represent the voltage; the plane formed by the end of the vector is in a so-called "voltage plane." This plane is conformal with the "current plane" formed by the end of the negative secondary current vector. Both these planes have a common point F_∞, which corresponds to infinite slip. Proof is given of a theorem which states that any change of the regulator voltage in the voltage plane and the corresponding change of the current in the current plane result in the same distortion when viewed from the common point F_∞. This theorem, which is claimed to be published here for the first time, is the main result of the whole paper. It provides a ready means for the graphical determination of the magnitude and phase of the voltage necessary to obtain minimum (active) current for a given slip and given load, and also gives curves of necessary additional capacitance reactance, so that the commutator motor may, under all conditions of regulation, be represented by an asynchronous circle. It is shown that this voltage can be conveniently resolved into three components, one of which corresponds to regulation, the second being proportional to no-load slip and the third to load-torque.

A simple graphical method for the evaluation of torques is derived, using the same scale for all conditions of regulation. Finally, exact and simplified equations for the calculation of no-load slip, compensation voltage, no-load current and speed drop are given.

OPEN HEARTH FURNACE PRACTICE

Novel Open Hearth Furnace Without Front Wall

By S. ICHIDA and D. I. BROWN. (From *The Iron Age, U.S.A.*, Vol. 26, No. 12, September 21, 1950, pp. 85-87, 3 illustrations.)

A NEW type of open hearth furnace, the H.I. furnace, named after the engineers, Hiraoka and Ichida, has been developed in Japan and introduced into practice with distinct advantages over the conventional type. The furnace was specifically designed with two objects in view: faster charging and longer life of roofs.

The furnace has a suspended roof and an all-door front eliminating the door jambs and arches in the furnace front wall. It has no front buckstays. The benefits of this type of design are claimed to be as follows:—

(1) Uniform and rapid charging is possible;
(2) on oversize scrap, 5 to 10 per cent of ingot cost has been saved through less scrap preparation;
(3) rapid bottom repairs are possible;
(4) no repair time of the front wall;
(5) scrap is closely packed, longitudinally, and dense, so that it heats and melts faster;
(6) roof can be partially repaired during operation;
(7) furnace productivity is increased by at least 20 per cent.

The H.I. furnace has only one door. As now used in Japan, it is a basic furnace of conventional design, except that it has no front buckstays and has a suspended roof. The frame across the front is made of heavy girders, rivetted, and water cooled. Initially, no oxygen was used, because none was available, but an oxygen generator will be built and used later.

Eventually, the furnace design will be modified to take advantage of the experience gained. This includes altering the construction of the front skewback, special protection of the wing wall bricks, and installation of a new charging machine.

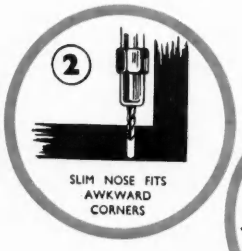
ORE TREATMENT

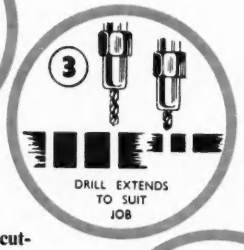
Hungarian Research Work on Improved Utilisation of Bauxites

By L. GILLEMOT. (From *Aluminium, Hungary*, Vol. 2, No. 2-3, March, 1950, pp. 25-32, 5 illustrations.)

THE processing of bauxites of high iron content will be economical only if these are utilised simultaneously for producing iron and alum-earth, and Hungarian research work has shown that this can be done. The bauxite is roasted and then a magnetic separation process is applied, as a result of which an iron-enriched part and a low-iron-content part are obtained. The iron-enriched part can be directly utilised as iron ore, whilst the low-iron-content part can be utilised to produce alum-earth according to the Bayer process. The drying process can be dispensed with, since the ore is roasted prior to magnetic separation. The iron-enriched part has a reduced content of slag components and is suitable for producing high-grade grey iron and steel for transformer sheet and welding rods. Titanium-oxide can also be reduced to obtain Ti, which is a valuable substitute for Ni in the production of stainless steel. Improved methods are being developed for production of alum-earth from the low-iron-content part remaining after the magnetic separation. Research work has been

5 ways to CUT DRILL BREAKAGE





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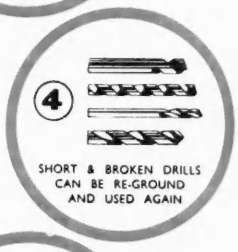

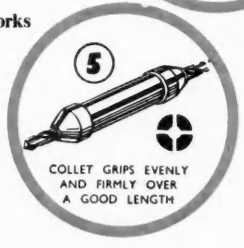

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done on the various problems concerning the simultaneous utilisation of bauxite as an iron ore and for producing alum-earth, but several of these problems have already been satisfactorily solved. Experiments on a production scale are under consideration.

SCIENTIFIC INSTRUMENTS

The Hot-Wire Anemometer in Supersonic Flow

By L. S. G. KOVÁSZNAY. (From *Journal of the Aeronautical Sciences, U.S.A.*, Vol. 17, No. 9, September, 1950, pp. 565-572, 584, 10 illustrations.)

THE hot-wire anemometer has proved to be a most valuable tool for turbulence research in the low-speed range. As interest grows in the problem of turbulence at supersonic velocities, it would appear natural to see whether the hot-wire anemometer could not also be used at higher speeds. However, a number of fundamental problems must be solved before the hot-wire can be considered as a successful instrument in the supersonic laboratory.

The first problem to be solved is that of obtaining adequate frequency response of the hot-wire equipment in order to resolve the turbulent fluctuations in space which are swept by the wire at high velocity in a supersonic flow. Encouraging progress has been made in using extremely fine wires and in developing electronic equipment that will compensate for the thermal lag of the wire at very high frequencies without interference from noise.

As yet, there is only a small amount of information available that can be used to predict the response of a hot-wire anemometer at such speeds. Because of the great difficulties of the theory of real compressible flows, most of the data will have to be obtained experimentally. The author has conducted systematic experiments to determine the law governing the heat loss from a hot-wire in a supersonic flow up to Mach Numbers of 2·0. It was found that the heat loss, expressed in the form of a Nusselt Number, is a unique function of the square root of the Reynolds Number when free-stream velocity and density are used and the viscosity and conductivity corresponds to stagnation conditions. This law is valid only in the case of small temperature loadings. For large temperature differences, a non-linearity appears in sharp contrast to King's law, which has been found to be linear with temperature difference over a wide range of temperature loadings.

The third problem which has been encountered is that of interpreting the hot-wire fluctuations when the number of flow parameters is large. Velocity, density, and temperature all contribute to the hot-wire reading, and it is believed that in some cases these effects may be separated. When the velocity field is a low-intensity turbulence (not random sound waves), the separation of measured fluctuations is accomplished.

THERMODYNAMICS

Report of Progress on Measurements of Friction Coefficients, Recovery Factors, and Heat-Transfer Coefficients for Supersonic Flow of Air in a Pipe

By J. KAYE, J. H. KEENAN and W. H. MCADAMS. (From Paper No. 50-F-13 presented to the American Society of Mechanical Engineers, September, 1950, 11 pages, 13 illustrations.)

MEASURED values are presented for friction coefficients, recovery factors, and heat-transfer coefficients for a stream of air flowing through a pipe at Mach numbers ranging from 2·5 to 1·2. The friction coefficients are in good agreement with those of Keenan and Neumann. At a diameter Reynolds number at inlet of about 1×10^5, which is intermediate between values observed by Keenan and Neumann, frictional effects of a very low order were observed. Maximum frictional effects occurred at a diameter Reynolds number at inlet of about $1{\cdot}5 \times 10^5$. Friction coefficients are, in general, considerably lower than those for fully developed turbulent flow of incompressible fluids at the same values of the Reynolds number.

For laminar boundary layer, it appears that the recovery factor is in close agreement with the results of the Pohlhausen analysis, namely, $(Pr)^{1/2}$ or 0·865. For turbulent boundary layer the recovery factor appears to vary from about 0·79 where the Mach number is 1·3 to about 0·865 where the Mach number is 2·2. It cannot be determined from the present tests whether this variation is primarily an effect of variation in Mach number, although it probably is.

The measured values of heat-transfer coefficient in the region of laminar boundary layer seem to be roughly in accord with the Pohlhausen analytical values for incompressible flow over a flat plate. In the region of greatest turbulence in the boundary layer, agreement with the McAdams correlation of data for turbulent flow in pipes is good. The major portion of the data obtained appears to be in the region of transition in the boundary layer from laminar to turbulent flow. Most of the observed values of heat-transfer coefficients are far smaller than the corresponding ones for fully developed turbulent flow—the ratio of one to the other being as small as 1 to 10.

WELDING

Optimum Flash-Welding Conditions for Aluminium Alloys

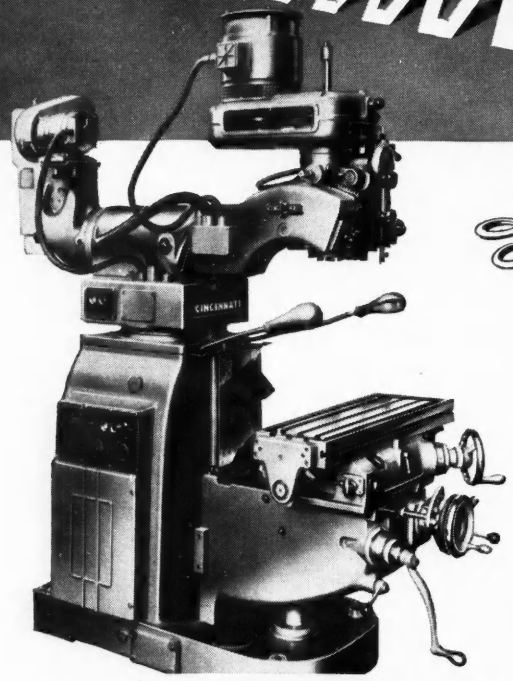
By E. F. NIPPES, W. F. SAVAGE, J. J. MCCARTHY and P. PATRIARCA. (From *The Welding Journal, Research Supplement, U.S.A.*, Vol. 29, No. 10, October, 1950, pp. 497-511, 36 illustrations.)

THE results of this investigation indicated that nearly 100 per cent tensile-joint efficiency could be attained by flash-welding various aluminium alloys. A comprehensive study of the peak temperature reached by the metal near the final weld line and the temperature distribution about the final weld line proved helpful in evaluating the optimum welding conditions for these alloys.

Welding variables which increased the peak temperature near the weld line and reduced the steepness of the temperature gradient were found markedly to influence the weld strength. The sensitivity of the alloys studied was found to differ. In general, faster flashing velocities, lower magnitudes of upset current, and shorter durations of upset current, all produce higher weld strengths.

The upset pressure requirements for producing various amounts of upset per specimen up to 1·5 times the section thickness were studied, and the influence of welding variables on these requirements was investigated. The magnitude of the upset current was found to be the most important variable influencing the upset pressure requirements, and the temperature distribution established during flashing was found to be of significance, particularly with the higher-strength alloys. In general, slower flashing velocities and larger magnitudes of upset current were found more effective in reducing the upset pressure requirements. It therefore became evident that a compromise in weld properties might have to be made in cases where the upset pressure available for welding larger sections would be the deciding factor. However, a significant reduction by an amount from 40 to 65 per cent in upset pressure requirements was found possible without reducing the tensile joint efficiency by more than 10 per cent, the greatest reductions in pressure requirements being associated with the stronger alloys. With one alloy, it was possible to permit selection of welding conditions so that a 50 per cent reduction in upset pressure could be realised without decrease in weld properties.

Precision duplication



The Cincinnati 8" x 18" Tool and Die Milling Machine was designed specifically for making dies and moulds quickly and accurately. The tracing head delicately scans the vertical profile of templates or full masters causing the spindle carrier to follow the same path automatically. Anti-friction screws are manually controlled and provide sensitive "feel" of the cutter action.

Universal mounting is provided for the spindle carrier so that it can be swivelled in two planes—an extremely useful feature when reproducing deep cavities, steep walls and sharp corners.

Intended for the production of small to medium sized work, specimens of typical work subjects are illustrated below to indicate how faithfully the master is reproduced and the very small amount of hard finishing necessary after machining.

Apart from duplicating work, such operations as conventional vertical milling, drilling, reaming and boring may be carried out. Useful attachments are also available such as—The Slotting Attachment which, when fitted as illustrated, can be brought over the table by swivelling the turret through 180° (and thus takine the spindle carrier to the rear) or the power driven Duplex Rotary Table Attachment, which turns the machine into an Automatic duplicator for some classes of work, one example being the glass mould shown below.

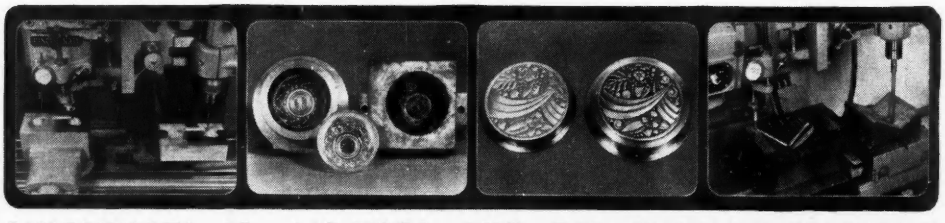
Further details of this machine and its application may be obtained on request.

Spindle carrier in vertical position making a plastic mould for telephone receiver.

Cast iron mould (left) made from composition master (right) for the glassware trade.

Steel embossing die made from composition master for medallion production.

Reverse image attachment is used to make a left-hand or right-hand die from the same master (golf club head).



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NEWS OF THE MONTH

PERSONAL

Mr. A. H. Bennett, general manager of Turbine Gears Ltd., Cheadle Heath, Stockport, has been appointed a director of the Company. **Mr. J. G. MacLean** and **Mr. N. Ramsay** have been appointed directors of Henry Simon (Engineering Works) Ltd.

Mr. Edward Burgess has been appointed managing director of Marshall Sons & Co. Ltd., Gainsborough.

Mr. W. F. Butler has been appointed a director of W. Sisson & Co. Ltd., Gloucester.

Mr. Robert Chalmers, O.B.E., B.Sc., M.I.C.E., M.I.Mech.E., has been elected President of the Institution of Engineers-in-Charge, St. Bride Institute, Bride Lane, Fleet Street, London, E.C.4. He has also been elected to be first President of The Engineer's Guild.

Mr. H. T. Chapman, general manager of Armstrong Siddeley Motors Ltd., has been appointed managing director of the Company. **Mr. W. F. Saxton, M.B.E.,** chief engineer and director, has been appointed technical and works director, and **Mr. W. H. Lindsey, M.A., F.R.Ae.S.,** has been appointed chief engineer of the Company.

Mr. H. Cunliffe has been appointed fuel-oil manager of Shell-Mex and B.P. Limited, Strand, W.C.2.

Major-General A. E. Davidson, C.B., D.S.O., M.I.Mech.E., M.I.Mar.E., has been elected President of the Diesel Engine Users Association, 56 Victoria Street, London, S.W.1.

Mr. Harold Drew, assistant chief engineer and director of Vauxhall Motors Ltd., Luton, Bedfordshire, has been appointed chief engineer in succession to **Mr. C. E. King** who will continue in general charge of the Engineering Department with the title of director of engineering.

Mr. F. J. Earnshaw, A.M.I.E.E., has been appointed a director of the Plessey Co. Ltd., Ilford.

Sir Lewis L. Fermor, O.B.E., D.Sc., A.R.S.M., F.R.S., has been elected President of the Institution of Mining and Metallurgy, Salisbury House, Finsbury Circus, London, E.C.2.

Mr. V. B. Harley-Mason, A.C.G.I., M.I.C.E., M.I.Mech.E., formerly with Tecalemit Ltd., has commenced practice as a consulting engineer at 316 Portswood Road, Southampton.

Mr. W. S. Hemp, M.A., A.F.R.Ae.S., has been appointed to the newly created Chair in Aircraft Structures and Aero-Elasticity in the Department of Aircraft Design at the College of Aeronautics, Cranfield, Bletchley, Buckinghamshire.

Mr. G. W. Heslett, chief estimating engineer of Sulzer Bros. (London) Ltd., 31 Bedford Square, London, W.C.1, has been appointed manager of the Pump Department of the Company.

Mr. B. P. Ingamells has been appointed director of merchant shipbuilding and repairs, Admiralty.

Professor R. O. Kapp, M.I.E.E., has joined Kennedy & Donkin, 12 Caxton Street, London, S.W.1, as consultant.

Mr. S. G. King, A.M.I.E.E., has been appointed London Manager of Birlec Ltd., with offices at 35 Park Street, London, W.1.

Mr. Henry Nimmo, M.I.C.E., M.I.Mech.E., M.I.E.E., Chairman of the Southern Electricity Board, has been elected Chairman of the General Council of The Engineers' Guild, 213 Abbey House, Victoria Street, London, S.W.1. **Mr. W. S. Graff-Baker, A.C.G.I., B.Sc., M.I.Mech.E.,** chief mechanical engineer (Railways) of the London Transport Executive, has been elected Vice-Chairman.

Mr. D. E. Proudlove, A.R.Ae.S., has been appointed civil air adviser to the United Kingdom High Commissioners for India and Pakistan.

Mr. J. B. Reed, B.Sc., A.R.I.C., has been appointed technical administrative officer of the British Coal Utilisation Research Association, Randalls Road, Leatherhead, Surrey.

Mr. L. C. Richards, B.Sc.(Tech.), M.I.E.E., has been appointed chief engineer of the Transformer Department of Metropolitan-Vickers Electrical Co. Ltd., Trafford Park, Manchester, in succession to **Mr. A. G. Ellis,** who now takes up the position of consultant to the department after 32 years as chief engineer.

Mr. D. M. Steel has been appointed chief technical engineer of the Renfrew works of Babcock & Wilcox Ltd.

Mr. J. Singleton-Green, M.Sc., A.M.I.C.E., has been appointed managing director of Stressed Concrete Design Ltd., 46 Great Marlborough Street, London, W.1.

Mr. G. E. Taylor, A.M.I.E.E., has been appointed technical investigation officer (electrical) to the London Transport Executive, 55 Broadway, London, S.W.1.

Mr. S. B. Warder, M.I.Mech.E., M.I.E.E., has been appointed chief officer (electrical engineering) to the Railway Executive Headquarters, in succession to **Mr. C. M. Cock** who has joined the English Electric Co. Ltd., Queen's House, Kingsway, London, W.C.2.

Mr. H. M. Woodhams, C.B.E., F.R.Ae.S., M.I.P.E., has been appointed managing director of Sir W. G. Armstrong Whitworth Aircraft Ltd., Coventry.

Mr. D. S. Woodley, A.M.I.Mech.E., technical director and chief engineer of Keith Blackman Ltd., Tottenham, N.17, has been appointed deputy chairman of the Company.

BUSINESS NOTES

A. C. Wickman Ltd., Coventry, announce their appointment as sole agents in Great Britain for the following Continental machine tool companies: **Gebr. Heller G.m.b.H.,** Nurtingen, for Cold Sawing Machines and Foundry Saws; Saw Blade Sharpening Machines; Milling Machines; Drilling and Boring Machines and Hydraulic Units; **Klopp,** Solingen Wald, for Mechanical and Hydraulic Shaping Machines; **Herlan & Co.,** Karlsruhe a/Rhein, for Cold Impact Extrusion Presses and Automatic Trimming Machines; **Collet & Engelhard,** Offenbach a/Main, for Horizontal Boring Machines and Die Sinking Machines; **Billeter Werkzeugmaschinen G.m.b.H.,** Bad Homburg, for Planing Machines; **Fritz Kopp,** Ulm/Donau, for Special Milling and Grinding Machines; Keyway and Slot Milling Machines and Copy Milling Machines; **Naxos Union,** Frankfurt, for Roll Grinding Machines; Crankshaft and Camshaft Grinding Machines; Surface Grinding Machines and Internal Grinding Machines; and **Lorenz A.G.,** Ettlingen-Baden, for Gear Shaping and Gear Hobbing Machines.

The equipment covered by these new agencies is generally available on reasonably short delivery.

Metalock (Britain) Ltd., 417 Grand Buildings, Trafalgar Square, London, S.W.1, announce that, following the recent tie-up with the Benelux countries, an office has been established in Legnano, near Milan, to carry out work for leading shipping and industrial firms.

Murex Ltd., Rainham, Essex, announce the formation of a new company known as Murex (Australasia) Pty. Ltd., with head office, factory and research department at Derwent Park, Hobart, Tasmania. Messrs. Francis H. Foster, C. P. Flockert and D. E. Hopkins are directors, and Mr. Wilfred T. Bennett general manager of the Company. Mr. E. J. Clarke, director and development engineer of Murex Welding Processes Ltd., is advising on the installation and operation of the new manufacturing plant.

Branch offices are to be established in all Australian State capitals and will be extended to New Zealand.

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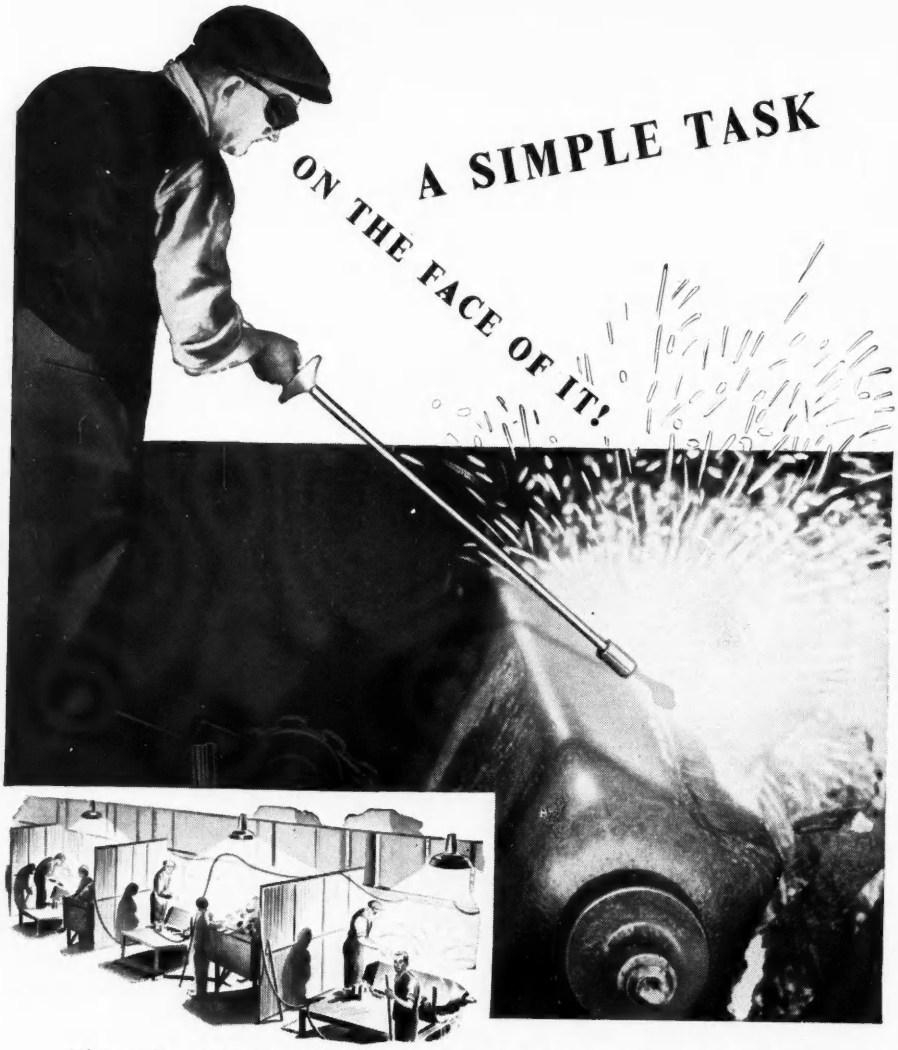


High quality steel products must be free from the defects which appear on the surface of the ingot, bloom, slab or billet during the process of steel manufacture. Oxy-Acetylene Hand Deseaming enables individual imperfections of any depth to be removed, either before or after the rolling operation, with a greater degree of precision than is possible with pneumatic chipping and in one tenth of the time—giving a faultless finish with an incalculable saving in man-hours and machine time. The deseaming of ingots illustrated above is just one of the many ways in which B.O.C. processes are aiding production in the Steel Industry !

THE BRITISH OXYGEN CO LTD LONDON AND BRANCHES

Lamberton & Co. Ltd., of Coatbridge, Scotland, have acquired the firm of forging equipment manufacturers known as Eumuco (England) Ltd., Beverley Works, London, S.W.13. Messrs. J. R. Lamberton (Chairman), W. A. J. Dinwoodie, F. Emanuel and A. R. McFadyen are now the directors of the Company.

May & Baker Ltd. have introduced "Planocop," a new high speed document copying developer, to meet the demand for a developer functioning within the range 30-60 seconds normally allowed for in the time cycle on which document copying machines operate.

"Planocop" document copying developer has been specially formulated to give images of good contrast and density with an exceptionally low fog level. The developing time can be fixed at any point between 20 and 120 seconds by suitable adjustment of the degree of dilution.

"Planocop" has a long working life, good storage properties and complete freedom from any tendency to stain or sludge.

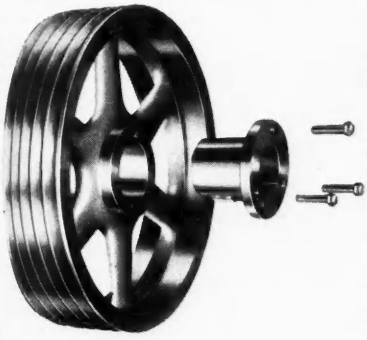
CROFTS ADAPTOR BUSH TYPE "V" ROPE PULLEY

Crofts (Engineers) Limited, Bradford, is the first Company to manufacture and market in this country the Adaptor Bush Type "V" Rope Pulley—a design which has been universally adopted and accepted in America.

This development is of much interest to machinery users and stockists, as the minimum number of pulleys used with selected Adaptor Bushes can be stocked to cover a maximum range of shaft diameters.

These Adaptor Bush Type "V" Groove Pulleys are *additional* to Crofts comprehensive range of "V" Groove Pulleys of normal construction which are in universal use. They are manufactured to cover a range of Standard "V" Groove Pulleys for $\frac{1}{2}$", $\frac{5}{8}$", $\frac{7}{8}$", and $1\frac{1}{4}$" section "V" Ropes, in a range of pitch circle diameters to provide speed ratios from 1-1 up to $7\frac{1}{2}$-1.

The Adaptor Bush has a horizontal slot milled through the length of the hub and flange, the action of pulling the Pulley on to the Bush by means of the three Setscrews has the dual effect of *Fixing both Pulley and Bush to Shaft.* Standard Feather Keys are used for transmitting power between Shaft and Bush, also between the Bush and Pulley.



To mount the Pulley on to its Shaft, all that is necessary is to assemble the Pulley on the Bush with the three Setscrews in position, and then to tighten them so that the shaft is gripped. To remove Pulley from Shaft, is equally simple.

The bushes are easily identifiable and are interchangeable in all pulleys of the same taper bore. The few simple operations required for mounting or removing the pulley can be carried out by any unskilled workman, the only tool required being the standard spanner.

Catalogues and details will be forwarded on request.

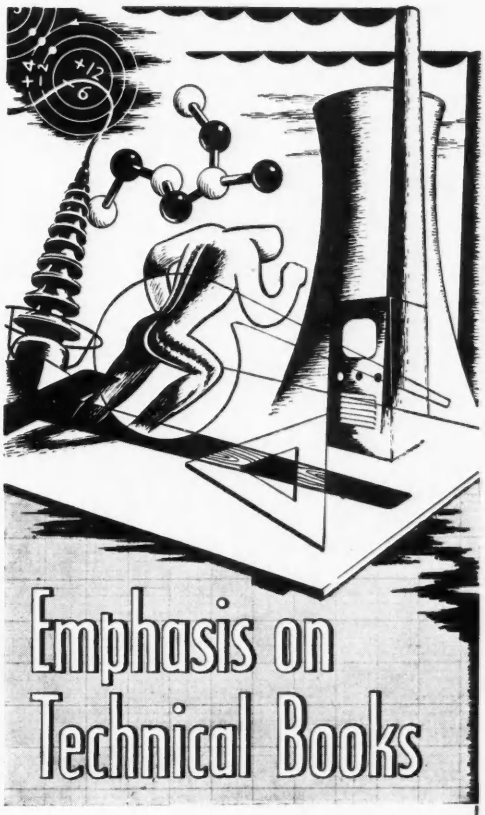
BRITAIN'S FIRST DIESEL ELECTRIC MIXED TRAFFIC LOCOMOTIVE

A new 827 hp diesel electric locomotive, No. 10,800, which has recently been completed at the North British Locomotive Company Limited's Works at Glasgow, is intended for secondary purposes, such as the operation of branch line and local train services, and will, together with the diesel-electric shunting and main line locomotives already in service, enable diesel traction to be investigated over the whole range of railway working.

No. 10,800 has been completed to the design and requirements of Mr. H. G. Ivatt, Chief Mechanical Engineer, London Midland Region, and is one of the projects sponsored by him and the former L.M.S. Railway Company.

The mechanical structure was detailed by the North British Locomotive Co. Ltd., and the whole locomotive built in their Works at Glasgow. The British Thomson-Houston Co. Ltd., Rugby, were the main contractors for the power equipment; the electrical generator, the traction motors and control apparatus were manufactured by them and they were responsible for designing the characteristics of the power equipment to suit the varied duties which it is intended shall be performed. The diesel engine was provided by Davey Paxman & Co. Ltd., Colchester.

The new locomotive is a double bogie design and has nose suspended traction motors mounted on each axle and driving through single reduction gearing. The general layout of the power equipment is similar to the standard 350 hp diesel electric shunting locomotives which have been so successfully operated for a number of years. A difference in this case, however, is the provision of a carriage warming boiler mounted in a separate compartment be-



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hind the driver's cab, and the re-positioning of the main fuel tank at the front of the locomotive with panel rediators on the bonnet sides, through which cooling air is drawn by a vertically disposed engine driven fan.

The power equipment consists of a Davey Paxman 16 cylinder R.P.H.X.L. engine rated at 827 BHP at 1250 rpm coupled to a British Thomson-Houston 6 pole 800 kw generator with a continuous current rating of 1600 A at 326 volts. The auxiliary generator and exciter are mounted above the main machine and chain driven from an extension of the main generator armature shaft.

The main control gear is located in a cabinet immediately in front of the driver's cab, which can only be opened by the master switch key, which in turn can only be withdrawn when the main circuit is isolated. The driver's cab has a centrally positioned control desk with all controls duplicated to permit driving rom either side. A vernier control is included for precise regulation of locomotive speed during hump shunting, or as required during normal operations. The control scheme allows for five engine speeds with load control provided to prevent overloading of the engine on all speeds except idling.

Compressed air is used for operation of certain contactors, the sanders, the horn and the re-fuelling equipment, but vacuum brakes are fitted to the locomotive, two 24 in. brake cylinders applying the brakes to all wheels through suitable linkage and slack adjusters. A deadman's pedal operating on the brake system is provided with a changeover cock to vary the action according to whether the locomotive is engaged on freight or passenger working.

The locomotive details follow a fairly conventional design, two four wheeled swing bolster bogies being used with S.K.F. roller bearing axleboxes and a combination of coil and laminated springs for suspension. The bogies are fitted with a spherical centre pivot which supports the weight of the locomotive and superstructure, and through which the tractive effort is passed. Side control pads are fitted to prevent undue oscillation of the main structure.

The power equipment is mounted on a very rigid sub-frame which is resiliently mounted on the two longitudinal " I " section beams which form the main frame of the locomotive. From these beams suitable fabricated gussets support the combined platform and sealing plate which runs the full length and width of the locomotive.

The maximum design speed is 70 mph, and the maximum tractive effort is 34,500 lbs with a continuous rating of 18,800 lbs at 12.3 mph, with a wheel diameter of 3 ft 6 in. Provision is made for propulsion starting of the main engine should the battery become discharged.

A comprehensive series of trials are being planned in order fully to explore the possibilities of a locomotive of this type and power for the varied duties which could be allocated to it.

HIGH SPEED HEAT

Modern jet fighters have become so hot when flying at high speeds that designers are faced with the problem of cooling them off. Cabin temperature rises not only with the heat of the jet engines : it also rises with the effects of skin friction between the air and the fuselage of the planes. Pilots often find their cabins uncomfortably hot especially when flying in the tropics.

The Royal Air Force have countered the problem by ordering a small refrigeration unit for cooling the cabins of their high speed fighters. The unit, which weighs only 9½ lb, works quite simply. It compresses hot air from the engine and then lets it expand so that it cools below the freezing point. This cool air then flows into the cockpit at a rate of something like 10 lb per minute.

The unit was developed by Sir George Godfrey & Partners Ltd., and has already been tested in a de Havilland *Vampire* 5 in Africa and the tropics.

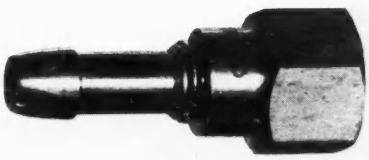
But the heat problem is still in its early stages. At speeds of about 600 miles-an-hour, the heat generated by skin friction is considerable. As the speed increases, so the heat increases. What the increase will be can easily be calculated in a rough way. Simply square the number of miles-an-hour. Thus at 700 miles-an-hour the temperature is 49° C (120.2° F). At 1,000 miles-an-hour the boiling point is reached. To touch a metal part at that speed would be like pouring scalding water over the hand.

Skin friction temperatures must be added to those of the air. If the temperature at a low altitude in the tropics is 40° C (102° F), the total heat of the aircraft is about 70° C (169° F). And even though the jet operates a few miles high, cabin temperature becomes considerably warmer than the air outside. At 20,000 ft, for example, where sub-freezing weather prevails, the temperature within the cabin may be 32.2° C (90° F).

A NEW SAFEGUARD FOR WELDING OPERATORS AND EQUIPMENT

Back fires may come about by one or a combination of circumstances, e.g., defective equipment, incorrect gas pressures, incorrect lighting up procedure, or careless handling of the blow pipe in use such as permitting the nozzle to touch the work, overheating the tip of the nozzle, or working with a loose nozzle.

In some cases a back-fire may pass beyond the injector and go back into either the oxygen or fuel gas hoses. It is then termed a " flash-back," and its effect is more serious in that it may result in immediate damage to hoses and regulators. There is also a possibility of injury to the operator. Warnings of a flash-back may be a squealing or hissing noise ; sparks coming out of the nozzle ; heavy black smoke ; or the blow-pipe handle may get hot.



Both blow-backs and flash-backs can be avoided by strictly following the instructions for the use of equipment. Over-familiarity can on occasion lead to neglect of the ordinary safeguards.

With this last point in mind, the British Oxygen Company are now providing an additional safeguard for operators. This safeguard takes the form of a special hose connector incorporating an anti-flash-back valve. The purpose being to arrest the flame at the inlet connections of the blow-pipe.

Essentially, the protector consists of a small non-return valve fitted into a hose connector of a standard size. It is constructed to operate without attention and will, in ordinary circumstances, have a life comparable with that of the rest of the equipment.

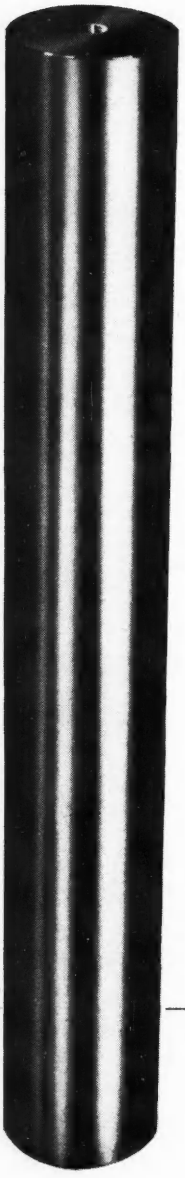
As implied, the B.O.C. anti-flash-back hose connector is fitted to the blowpipe end of the oxygen and fuel gas hoses only, and a standard hose connector will still be used for connecting these hoses to their respective regulators. As the device is essentially a non-return valve, it will be apparent that reversal of the hoses will prevent the flow of gas to the blowpipe.

All standard welding and cutting blowpipes will, in future, be supplied by the Company with lengths of hose and connectors already fitted with these safety devices.

Prolonged practical tests in workshops with the new B.O.C. device show that it is very effective and it is recommended to all users of gas welding equipment. Replacement of existing hose connections with the safety type can be effected in a few minutes.

INTERNATIONAL WELDING CONGRESS

An International Welding Congress is meeting in London and Oxford from 14-21 July next year. The Congress is being sponsored by the Institute of Welding, the British Welding Research Association, the British Acetylene Association, the Welding Sections



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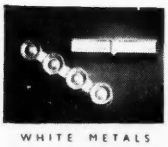

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of the British Electrical and Allied Manufacturers' Association and the Sheet and Strip Metal Users' Technical Association. The President of the Reception Committee is Sir William Larke, K.B.E.

The Congress will open in London on Saturday, 14 July, and will be transferred on the following day to Oxford, where three colleges, Christ Church, Oriel and Somerville, will accommodate the visitors, who are expected to number between four and five hundred.

Meetings of the fourteen technical Commissions of the Institute will occupy the first two days, and in the second part of the week there will be three open sessions for the presentation and discussion of papers as follows: The Welding of the Wrought Light Alloys, Chairman: Sir Arthur Smout; The Welding of Bridges and Allied Structures, Chairman: Professor J. F. Baker (Cambridge University); Present Trends in British Welding (Chairman not yet appointed).

The Secretary of the Congress is the Secretary-General of the International Institute of Welding, Mr. G. Parsloe, and the office of the Organizing Committee is at 2 Buckingham Palace Gardens, Buckingham Palace Road, London, S.W.1.

The largest Blast Furnace in Great Britain, having a hearth diameter of 25 ft 9 in. and rated to produce 1,000 tons of basic iron per day, was blown in at the Margam works of the Steel Company of Wales on November 7th. It forms one of the group which will supply hot metal to the steel plant at the Abbey Works where the largest hot strip mill in Europe is now being installed. The new furnace, like the other at Margam, was designed and built by Ashmore, Benson, Pease & Co. Ltd., of Stockton-on-Tees.

Classified Advertisements.

The rate for all classified advertisements is 6d. per word; in bold print 9d. per word; minimum order 6s. Box number advertisements 1s. extra. Instructions together with remittance must be received not later than the 3rd of each month for advertisements to appear in the same month's issue.

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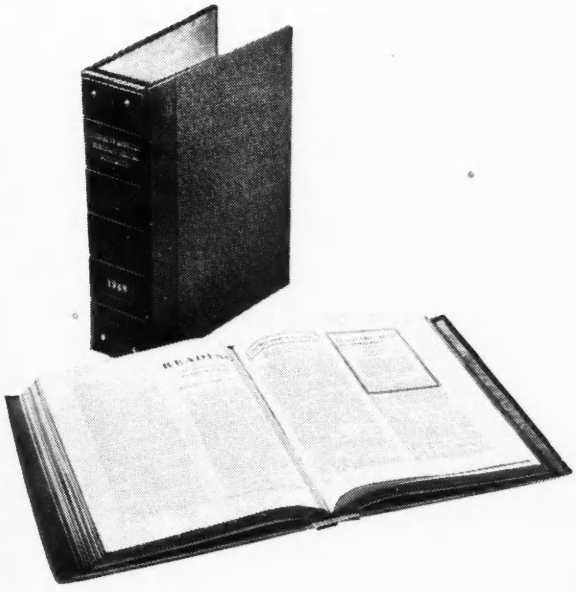
GAS-SAMPLING APPARATUS

(*Concluded from page* 382.)

there is no leakage. For detailed investigations, it is necessary to make exploratory tests at various positions. To make it easier to locate the positions at which leakages occur, a small removable funnel covered with a slotted plate may be fitted over the fan housing and directed towards the position to be investigated.

Tests can also be made with this apparatus to determine whether personnel working with substances of the types mentioned are apt to absorb any appreciable quantities of these substances either internally or in their clothing. The tests should be carried out at a place where the air itself does not produce any reaction. The persons being examined should take a deep breath and then breathe out slowly into the intake side of the apparatus, which is held in such a way that the air exhaled will not cause appreciable air currents in the catalyser tube. The air from a single respiration is then sufficient to indicate whether the person examined shows any symptoms of poisoning.



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THE BLUE PAGES

A QUICK GLANCE AT THE LATEST TECHNICAL DEVELOPMENTS THROUGHOUT THE WORLD

● *ELECTRONIC FLOW METER*

A new type of electronic flowmeter incorporates a novel flow-sensing rotor which spins freely within a venturi. According to the claims of its makers, this new design avoids problems of bearing maintenance, friction drag, and high pressure drop such as is associated with previous bearing-mounted turbine-type rotors. Fluid passing through the non-magnetic housing turns the rotor at a speed determined by the velocity of the liquid and pitch of the rotor blades. A cylindrical permanent Alnico magnet within the dynamically balanced rotor sets up a rotating field which acts upon an external pick-up coil. The electrical impulses produced in the latter are conducted to a combination electronic integrator and totalizer. The integrator transforms the current from a.c. to d.c., the magnitude of which is directly proportional to the rate of pulses or frequency of the current. Another circuit then converts the current into short d.c. pulses which are recorded on the totalizer. Since the recorded pulses are proportional to the rotation of the sensing rotor, the flow rate and total volume of fluid can be read directly in volumetric units, or, if fluid density is constant, in gravimetric units. The device can be used to measure the flow of any fluid, it is claimed, and operating temperatures are stated to range from 4° absolute to 1,000° F, while standard designs are suitable for pressures up to 5,000 psi and special housings are built for pressures up to 20,000 psi.

● *TENSILE STRENGTH AS CRITERION OF CUTTING PERFORMANCE*

Tool life, i.e., the time a tool remains serviceable under a given set of conditions, is probably the most important practical index of machinability. An appreciable amount of data is now available, which can be expressed by the formula $VT^n = C$, where V is cutting velocity in feet per minute, T is tool life in minutes, and n and C are constants depending on the cutting conditions, nature of work material, and other factors. While this relationship between velocity and tool life is quite general, the constants n and C are applicable only to specific conditions. For each new condition of metal cutting on which no information is available, time-consuming and costly experiments must be carried out in order to find the numerical values of these factors. Consequently, a correlation of tool life with physical constants pertaining to the cutting conditions, similar to the correlation of the friction factor with the Reynolds number in hydrodynamics, would be a valuable contribution. It is, therefore, important to note that a recent investigation of the metal cutting process on the basis of the plasticity theory has shown that the state of plastic deformation of metal cutting can be correlated with that of an ordinary tensile test. This means that the tensile properties of a material may offer a useful index of cutting performance. Considering the great importance of this matter, no time should be lost in investigating the possibility of evolving a technique for the practical application of this method.

● *SURFACE HARDENING OF MILD STEEL*

In electromagnetic gear boxes of the type in which the armatures perform the function of clutch plates, the mechanical softness of mild steel makes this material unsuitable to withstand the wear involved. In order to retain both the magnetic and the adhesive properties of mild steel, a special nitriding treatment has now been developed. The new process involves the use of a mild steel material which contains no more than 0·1 per cent carbon and a certain percentage of aluminium, the latter not to exceed 0·02 per cent. When subjecting this material to nitrogenation, a surface hardness of 200 Brinell results, which is about one-third that of the ordinary quench-hardened nitrided steel. Depth of penetration achieved with the new process, however, is about ten times greater than that obtained with ordinary nitriding steels, which contain higher percentages of carbon and aluminium and also contain vanadium and chromium in percentages ranging from 0·5 to 2 per cent. The hardening with nitrogen can be effected by one of the usual nitrogenation or cyanidation processes, or by merely lapping the friction surface of the clutch plates. This lapping produces frictional heat and forms atomic nitrogen, which is absorbed by the surfaces involved. This constitutes a special advantage, since the hardened layer is self-renewing as wear takes place, merely as a result of the friction produced by operation of the clutch. There should be many other applications where the new process can be applied with considerable advantage.

MUREX PRESENTS

'HARD METAL'

(Produced by The Gateway Film Productions)

'Hard Metal'—a colour sound film of approximately 45 minutes' run—shows in vivid detail some of the manufacturing processes involved in the quantity production of Cemented Tungsten Carbide. Later stages of the film show 'Prolite' Tools, Dies and Wear-resisting products at work in industry and give some idea of the wide field of application for this interesting material 'Hard Metal'. The film will shortly be shown throughout the country and personal invitations are now being despatched by Protolite Ltd., the sales organisation for 'Prolite' products.

CEMENTED TUNGSTEN CARBIDE

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● *ULTRASONIC SOLDERING OF ALUMINIUM*

The successful introduction of a soldering iron employing ultrasonic vibrations has done much to focus attention upon the utilisation of such vibrations for similar purposes. It has been found that the surface of aluminium to which molten solder is applied can be " tinned " by exposing it to the action of ultrasonic vibration. However, this tinning does not occur if it is attempted to carry the process out under pressure, which suggests that the mechanism involved is one of cavitation erosion of the oxide film. In accordance with the nature of the cavitation mechanism, ultrasonic tinning is less effective at high frequencies, while it occurs quite readily at lower frequencies with the employment of a relatively low intensity of vibration. This observation appears to be confirmed by experience acquired in the application of a new cast-welding technique, in which molten aluminium is poured into a mould to form a joint between parts which have to be joined together. The molten aluminium is allowed to flow through the mould for some time until the oxide films present are broken up by the application of vibrations and are carried away by the stream of molten metal. When this is achieved, the flow is stopped and the metal allowed to solidify, thus making the joint.

● *NEW METHOD OF BRAZING AND SOLDERING*

The speed of soldering and brazing on a production basis can be greatly increased by employing a novel mechanised method to be used in connection with a suitable conveyor system. In this method, brazing or soldering is carried out with the employment of a so-called " gun," which feeds the brazing or soldering wire at some fixed speed which synchronizes with the speed of movement of the workpiece being processed. The speed of the wire as it passes through the gun can be varied to meet the requirements at hand ; but, once adjusted to the required speed, the wire maintains that speed indefinitely, and brazing or soldering becomes a continuous operation. As a rule, the gun remains in a fixed position during the operation, but it is also possible to move the gun and keep the work-piece stationary. Alternatively both the workpiece and the gun may be made to move. When set up for brazing, the gun is usually mounted over the workpiece so that the latter will pass along under the gun. The brazing tip is set a fixed distance away from the surface to be brazed. The speed at which a seam can be brazed will depend upon the amount of copper required to fill the seam, the diameter of the wire used, and the gauge and material being worked on. In practice, it has been found that seams can be brazed at the rate of 60 to 120 lineal inches per minute.

● *IMPROVED NAILS*

Directly or indirectly, nails are in everyday use by almost everyone. This fact contrasts strangely with the lack of knowledge concerning the behaviour of nails, once driven into wood. Actually, experience alone governs the use of nails. If one nail does not perform the task, a second one is used. If a large nail splits a piece of wood, one or two smaller ones are used instead. Based upon requirements and individual preferences of the various nail-using trades, the number of types and sizes of nails in use and production is exceedingly great. In a recent investigation, no fewer than 3,600 tests were performed on the driving resistance, withdrawal resistance, lateral load carrying capacity, and deformation of 2 to 4 inch-long plain-shank, spirally grooved, and annularly grooved nails in various woods, in order to obtain comparative test and design information. An especially interesting and significant observation relating to the performance of heat-treated high-carbon steel nails was that the withdrawal resistance and lateral load carrying capacity can be considerably increased by hardening, and this without affecting driving resistance.

● *TRANSFORMER OIL RESEARCH*

Depending on their composition, transformer oils, when exposed to the influence of an electric field of sufficiently high intensity, may become capable of either absorbing or giving off gases. This phenomenon is of considerable importance in high voltage transformer engineering, where the properties of the oil must be such as to absorb any gases present and not to release gases, as the latter would lead to the ultimate breakdown of the apparatus. In order to be able to judge transformer oils from this aspect, it is necessary to have available a reliable and sufficiently sensitive apparatus for testing the oil as to gas release or absorption.

The possibility of developing transformer oils with improved characteristics as regards gas absorption must, of course, rest upon a full knowledge of the specific chemical and or physical properties of the oil which are responsible for gas release and absorption respectively. A number of investigations carried out in recent years have shed some light upon this problem. A recent report on experimental data obtained with the use of improved test equipment renders important information on such influences as frequency of the testing current, electric field strength, oil temperature, etc. Based upon these findings and upon certain theoretical considerations, a fairly complete theory of the phenomena involved is given. It is hoped that this latest contribution to the knowledge of transformer oil characteristics may stimulate further efforts to produce oils of improved quality.

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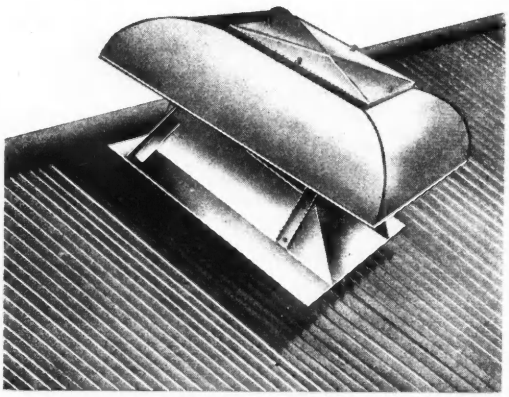

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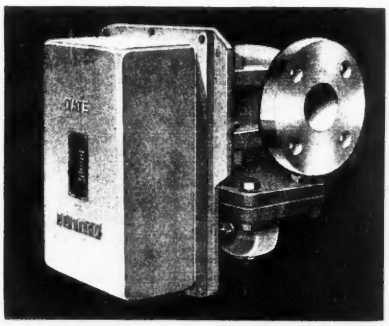

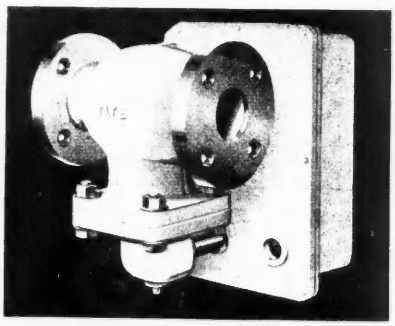

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● *NEW WELDING PROCESS*

In order to increase welding speed in manual arc-welding, a method has been evolved and perfected, in which two welding rods are simultaneously employed. The one electrode is handled by the operator in the usual manner, while the other, which is mechanically supported by a guiding device, rests with its insulated coating upon the welding seam in a manner similar to that employed in contact arc-welding. The welding operation is thus carried out with the simultaneous establishment of two arcs between workpiece and electrodes. It is reported that the weld seams produced by the new method are of the same quality as, if not better than, welds produced by orthodox manual welding A great economic advantage of the new process is that the rate of deposition of weld metal is twice that of the ordinary process, and this rate can be further increased by using a thicker rod for the mechanically guided electrode. Application of this method is said not to be confined to mass production ; it does, however, require the use of welding rods specially adapted to the process.

● *THE PHOTOTRANSISTOR*

Experiments with transistors of the germanium type have revealed that the exciting signal can be produced by the absorption of light. This observation has led to the development of a new photoconductivity cell, the " phototransistor." The heart of this device is a pill-shaped wafer of germanium having a spherical " dimple " ground on one side so that the thickness of the wafer at the centre is about 0·003 inch. The wafer is secured at its periphery in a retaining ring serving as electrode. A pointed wire electrode, the " collector," of 0·005 inch phosphor bronze wire bears upon the germanium at the centre of the dimple. In a direct-current circuit in which the phototransistor is connected in series with a battery and load resistance, a small current will flow even when the photocell is in the dark. When light falls on the germanium, the current is increased in proportion to the amount of light falling upon the cell. One of the distinguishing features of the phototransistor is that a substantial photocurrent response is obtained only when light falls in the immediate neighbourhood of the collector on the surface of the germanium. This cell is especially suitable with incandescent light sources.

● *IMPROVING INTERNAL COMBUSTION ENGINE EFFICIENCY*

Maximum efficiency of an internal combustion engine will obtain if the products of combustion are expanded to a point where their temperature equals that of the surrounding atmosphere. It is well known that the actual performance of engines is very far removed from this ideal process. Theoretical consideration of the problem points to a solution in which the waste heat of the products of combustion exhausted by the engine is imparted to compressed air by the employment of a heat exchanger, the heated air being subsequently expanded in an air engine. Such a process could be realised by combining an internal combustion engine with an air compressor, a heat exchanger and an air engine. By employing a closed air circuit, in which the pressure of the air after expansion in the air engine still lies above atmospheric pressure, the advantage of reduced compressor and air engine dimensions would be obtained. According to a recent proposal, air compression and expansion can be carried out in one and the same cylinder, and the latter may be made to form part and parcel of the internal combustion engine proper. An engine of this type would then consist, e.g., of four internal combustion cylinders and one air cylinder. It is estimated that in the case of a gas engine, such an arrangement would produce a 25 per cent increase both in engine efficiency and engine power.

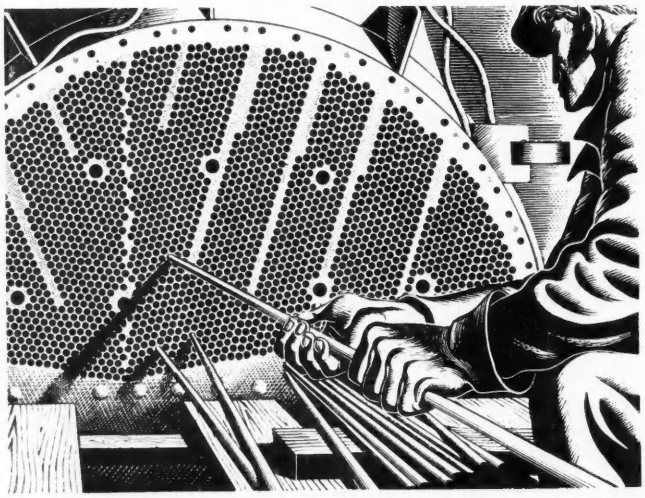
● *IMPROVING ACID RESISTANCE OF ENAMELS*

Most porcelain-enamelled articles are exposed in some degree to abrasion during normal use, and acid attack is also frequently present. During an investigation into the relative effect of certain acids on porcelain enamel, some peculiar effects were noted, in which the resistance of a number of enamels to solution in citric acid was considerably improved by pre-treatment with certain weak organic acids which did not themselves appreciably attack the enamel. Upon closer investigation, it was found that although pre-treatment with 10 per cent acetic acid or 2 per cent butyric acid for 15 minutes at room temperature produced little or no visible attack on some enamels, it did inhibit further attack when the enamels were subsequently treated in 10 per cent citric acid. Acetic acid in concentrations of 0·5 to 50 per cent was found to produce the passivating effect. The effectiveness of treatment with 10 per cent acetic acid increased with the time of treatment up to 5 minutes, after which no increase was noted. Treatment with 10 per cent citric or 10 per cent acetic acid markedly decreased the abrasive resistance of some enamels, the effect of citric acid being much greater than that of acetic acid. Titanium-opacified enamels were found to be particularly sensitive to these effects.

● *ELECTRONIC INSPECTION OF ENGINE CAM CONTOURS*

The camshaft lobes which operate the valves of internal combustion engines must be accurately ground if good engine performance is to be achieved. The conventional method of cam inspection by recording cam contour lift at intervals of one or two degrees is time-consuming

NON-FERROUS PLATE and condensers



Tube plates fitted to condensers in marine and land installations must offer high resistance to corrosion and erosion. The Metals Division of I.C.I. manufactures condenser plates to customers' requirements in Naval Brass, Muntz Metal, 70/30 Brass, Aluminium Brass and Cupro-nickel, in accordance with any of the British, American or other recognised standards.

The consistently high demand for I.C.I. condenser plates and tubes is ample evidence of their quality and reliability—the outcome of many years' production experience, coupled with rigid technical control. The resources of the Division are freely available to help in the solution of technical problems concerning the use of such materials.

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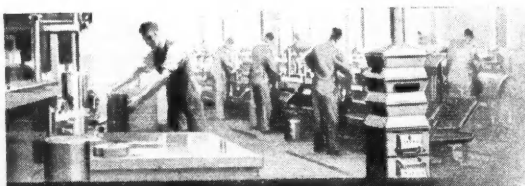

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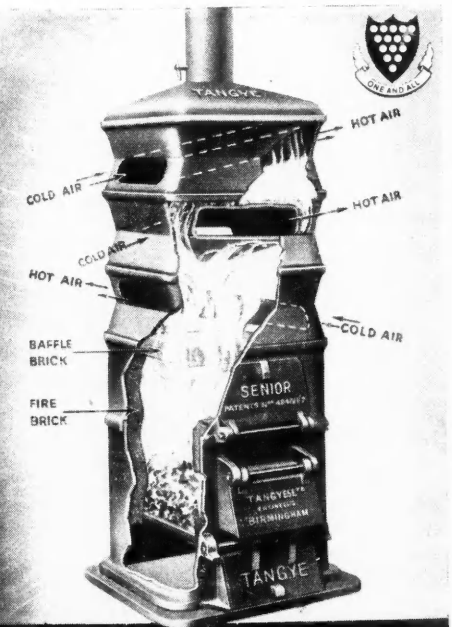

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and unsuitable for routine inspection. The template method using a magnifying comparator is accurate, but still does not lend itself to production inspection. By using the recently evolved electronic method, it is possible to inspect one lobe per minute. In this method, the camshaft is mounted in a lathe and each cam contour range is checked by observing curves on the screen of a cathode-ray oscilloscope. By turning a three-position switch, this equipment produces on the screen either lift, velocity, or acceleration curves. Of these, the acceleration curve is best of all for critical cam contour inspection, as it reveals irregularities even better than by calculation from lift figures measured to one-tenthousandth inch. The tappet follower used in inspection work should be the same as the type normally used with the cam contour being inspected. The same type of equipment may be used to observe the motion of i.c. engine valves at high speed.

● *SMALL GRINDING MILLS*

A vibratory ball mill recently developed at the National Bureau of Standards, Washington, is reported to grind cotton rapidly and completely into particles a few microns in length, with little or no oxidative change or contamination of the cellulose. Grinding takes place as 3700 steel balls, colliding over 100,000 times per second, pulverise the material caught between them. As the new mill produces a more uniform powder and is more efficient and easier to operate than previous devices of this kind, it is expected to find extensive application in the grinding and blending of a variety of substances, such as pigments, metal powders, resins and plant and animal tissues. This mill closely resembles the small vibratory ball mill referred to in the Blue Pages of the September issue (p. 300). The need for mills for fine grinding is emphasized by yet another report concerning the development of a small beater-type mill. This mill, which is built on a commercial scale in several capacities, is of the high-speed beater type, in which the material to be comminuted is introduced into pockets surrounding the beater wheel, and the arrangement is such that the material cannot pass out of the containing pockets until it is reduced to the specified fineness. The latter is adjustable by means of a control lever during operation.

● *ELECTRONIC QUENCH CONTROLLER FOR STEEL*

A recently developed electronic apparatus for controlling the quench time in cooling heat-treated steel parts is based upon the fact that a martensitic steel structure is ferromagnetic, while the austenitic structure formed at high temperatures is not. Up to the present, the best available instrument for controlling the quench has been a clock, but this method has several drawbacks arising from variations in the quenching characteristics of the material, and size and shape of the steel part to be hardened, and from variations in the initial temperature of the part, the amount of scale present on its surface, and the temperature, agitation and impurities content of the water in which quenching is carried out. The heart of the controller is a system of coils which is placed in the quench tank. This system consists of a centrally located driver coil which excites two identical pick-up coils, one above the other, at identical distances from the driver coil. The pick-up coils are electrically balanced, but when a ferromagnetic piece is introduced into one of the coils, the initial balance is disturbed and a relay is operated which energizes a bell and lights a lamp, indicating that the steel part should be withdrawn from the tank. In operation, this will happen when the piece to be hardened is introduced into the water space surrounded by the upper coil and ferromagnetic martensite begins to form because of the quench.

● *TRIPLE-LAYER CORROSION RESISTANT PAINT*

It has been found that a mica-base three-coat paint will considerably lengthen the service life of outdoor transformer tanks exposed to seacoast and industrial atmospheres. This so-called " coastal finish " is applied in the same manner as standard finishes. In production, it is baked-on for speed, but a modified air-drying finish permits patching of damaged surfaces in the field. The prime or first coat applied to the prepared surface is composed of a vehicle consisting of a combination of modified phenolic and alkyd resins. The pigments used are primarily zinc chromate and iron oxide. The second or intermediate coat is the key to the successful performance of the first. Its pigment is composed of selected mica flakes which overlap each other in the film, thus producing a shingle roof effect. Mica also increases the heat stability of the intermediate coat as much as ten times at elevated temperature. The vehicle used for applying the mica pigment is composed of modified phenolic and thermoset type alkyd resins. The use of mica in paint is not, of course, new, but it is claimed that this particular development solves the difficulties involved in the employment of relatively large mica plates. The final coat is composed of resins and pigments selected for the conditions involved. With ingredients slightly changed to fit other requirements, the process could be used on many other surfaces, possibly even as an undercoating on automobiles.

For further information on developments reviewed in THE BLUE PAGES, readers are invited to write to the Editor.

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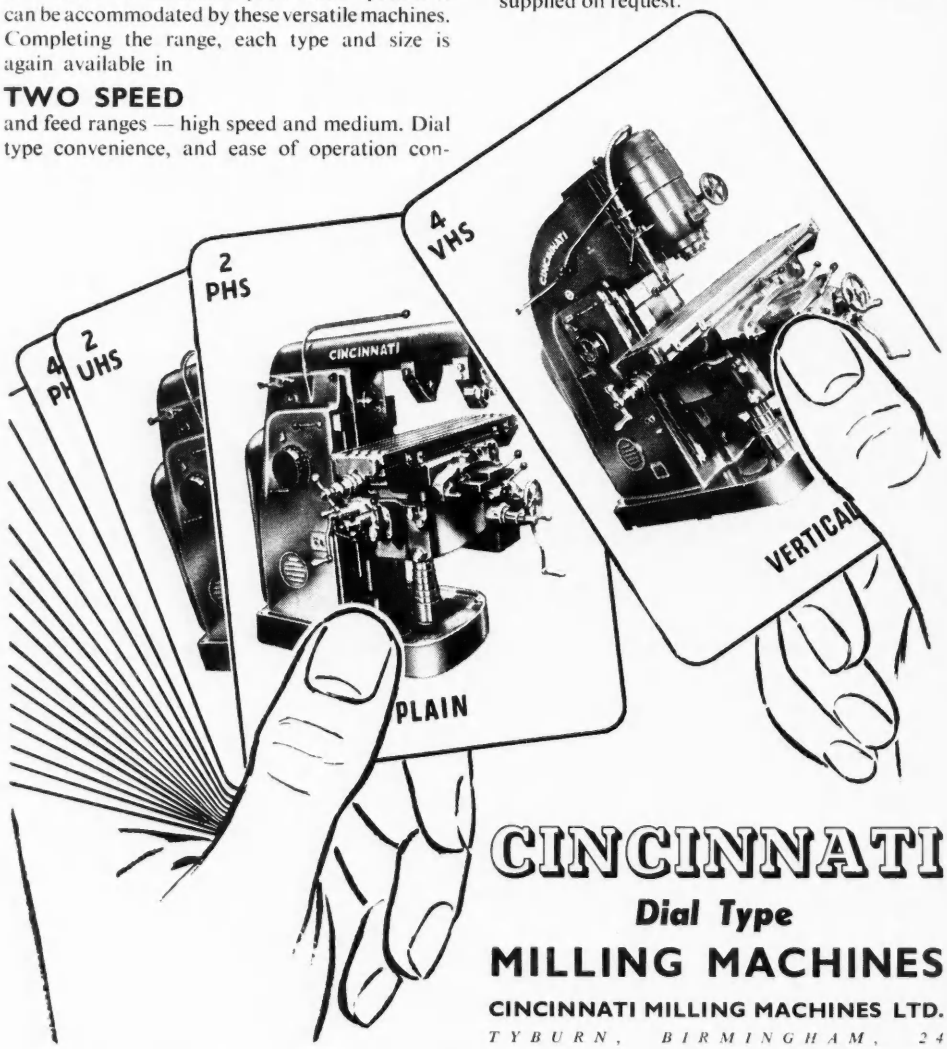

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